MACHINE DESIGN

July 1946

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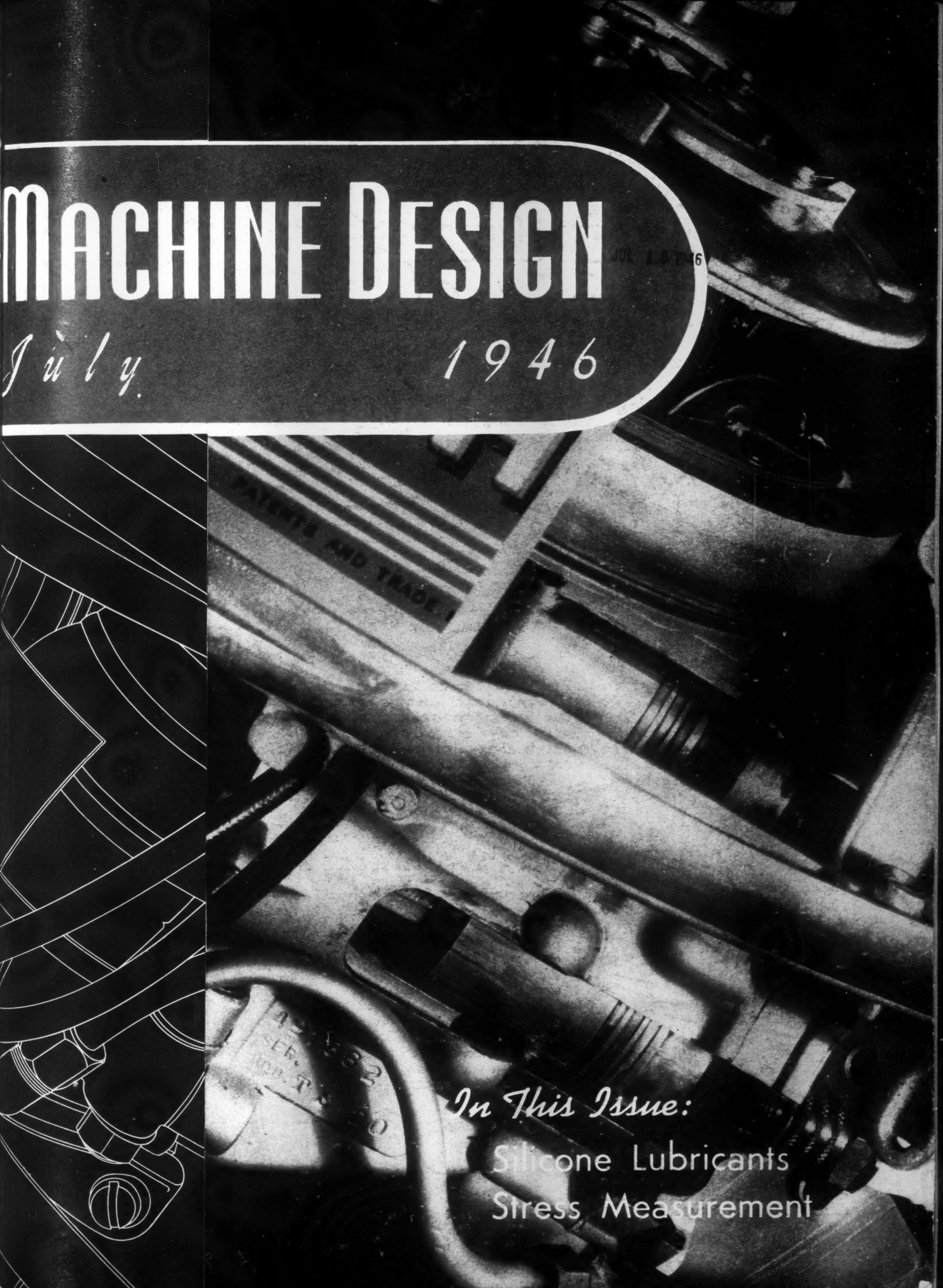
PATENTS AND TRADE
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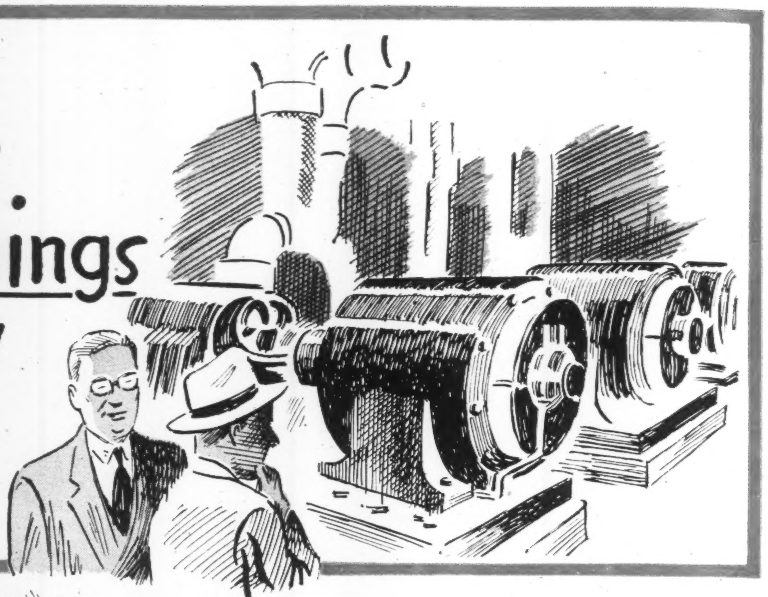
Silicone Lubricants

Stress Measurement

"For This Job, Give Me Two Things in a Motor"

. . . said the Power Plant Engineer



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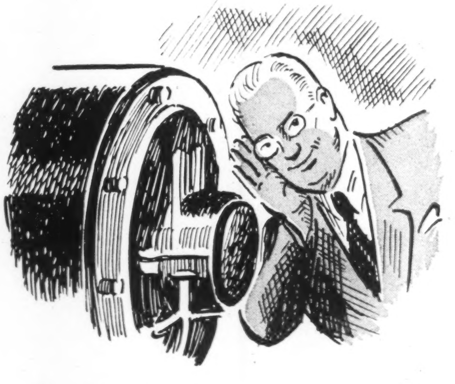

Frankly, Mr. Reader, we *are* mighty proud of our 2-pole motors. We've been developing them for high-speed requirements like this from the very earliest. Licked a lot of problems too. Like cutting down noise through better design . . . reducing windage noise with proper baffling, and curbing magnetic noise . . .

We're Getting Better Efficiencies and power factor . . . and lower starting kva, through improved squirrel-cage design. We've designed bearing brackets generously for greater structural rigidity. In fact, all the way around you can expect to find long life and low maintenance built-into our motors!

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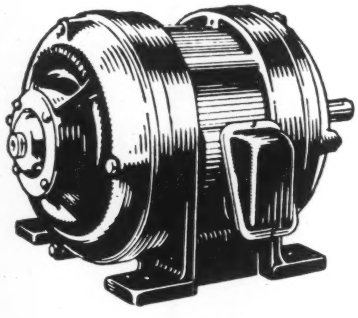
A 2082



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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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MACHINE DESIGN
July 1946

THIS MONTH'S COVER: Cutaway view of Johnson Seahorse outboard motor. Developing five horsepower at 4000 rpm the engine is a twin-cylinder, two-cycle type. Weighing only 44 pounds the engine propels small boats at 10 miles per hour

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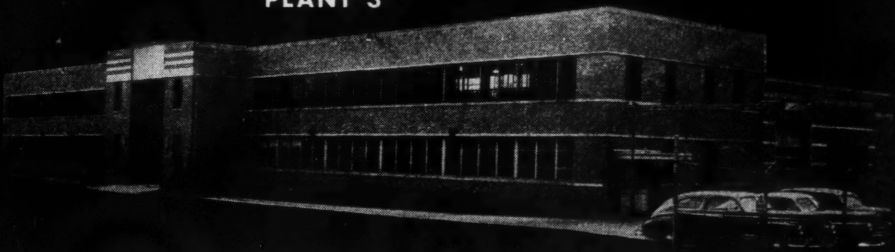
PLANT 4

PLANT 7

PLANT 6



PLANT 3

PLANT 2

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Topics

CERAMIC AND SINTERED MATERIALS have been developed by the Germans for use in the production of high-heat resisting turbine blades, according to a report released by the Office of the Publication Board, Department of Commerce.

WITHOUT BENEFIT of brush or spray gun, a new paint may be applied by wiping on with a piece of cheese cloth. Known as "Dab" the product has been developed by the Automotive Division of Motel Supplies.

PRIVATELY-OWNED AIRPLANES should be equipped with tricycle landing gear and automatic flap, be stallproof and spinproof, have a 50 mile-an-hour minimum flying speed, and incorporate design simplifications, according to J. M. Gwinn Jr. of Consolidated Vultee Aircraft Corp.

FLAME SPRAYING process has been developed for applying plastic coatings to materials. Perfected by the Powder Weld Co. it can also be used to weld or braze without the use of rod, wire or separate flux. Mechanical mixtures of powder metals and suitable fluxes are projected through the flame cone into the work surface.

STEERING of tomorrow's light passenger cars may be accomplished by two balanced hydraulic circuits, according to the B. F. Goodrich Co. One circuit will effect a hydraulic tie rod between the front wheels, the other will be a pump circuit. Both will be kept under minimum pressure by a spring-loaded reservoir. Also, a hydraulic constant-level mechanism, applied to all four wheels will maintain chassis height regardless of wheel movements.

PHOTOGRAPHIC STUDIES of how a lubricant acts inside an internal-combustion engine has provided engineers with information on the performance of pistons, rings and oil film. Using glass cylinders and camera, M. C. Shaw and T. J. Nusdorfer of the National Advisory Committee for Aeronautics revealed that piston-skirt lubrication is hydrodynamic for at least part of the stroke and that the amount of lubrication is a function of the relative angular positions of the piston rings and the lateral motion of the piston. The rings were observed to rotate as much as one revolution per minute at engine speeds of 1000 rpm.

STANDARD GRAYS recommended by ASA have been selected jointly by Westinghouse and General Electric to achieve color harmony for separate products grouped together or for two-tone treatment of a single unit. These grays, identified by the Munsell-spectrophotometer system, are light gray, 2.5 GY 6.0/0.5; medium light gray, 7.5 BG 4.5/1.0; medium dark gray, 6.5 B 3.3/1.0; and dark gray, 10.0 B 2.5/1.0.

TO ENCOURAGE study and preparation of papers on design, research and education with respect to the application and use of arc welding, the James F. Lincoln Arc Welding Foundation has announced a new $200,000 award program. Details may be obtained from the secretary of the foundation in Cleveland.

DIAPHRAGMS OF FIBER GLASS for sensitive control of instruments have proved superior to goat's skin, according to the Askania Regulator Co. The new diaphragms are nonporous, do not stretch, and are impervious to gases and vapors. They also have high resistance to alkalis and acids and have much longer life than goat leather.

ROCKET MOTORS can get the same mileage at 2750 mph that a typical reciprocating aircraft engine with propeller gets at 350 mph from a pound of propellant mixture, both operating at an efficiency of 25 per cent. It is reasonable to assume, however, that greater efficiencies than the present maximum of 28 per cent may be obtained from rocket power in the near future.

Endless Belt Recorder

Design features include mechanism which handles 0.005-inch thick plastic belt, and drive system that insures constant recording speed

By F. W. Roberts
and A. W. Skoog
Dictaphone Corp.
Bridgeport, Conn.

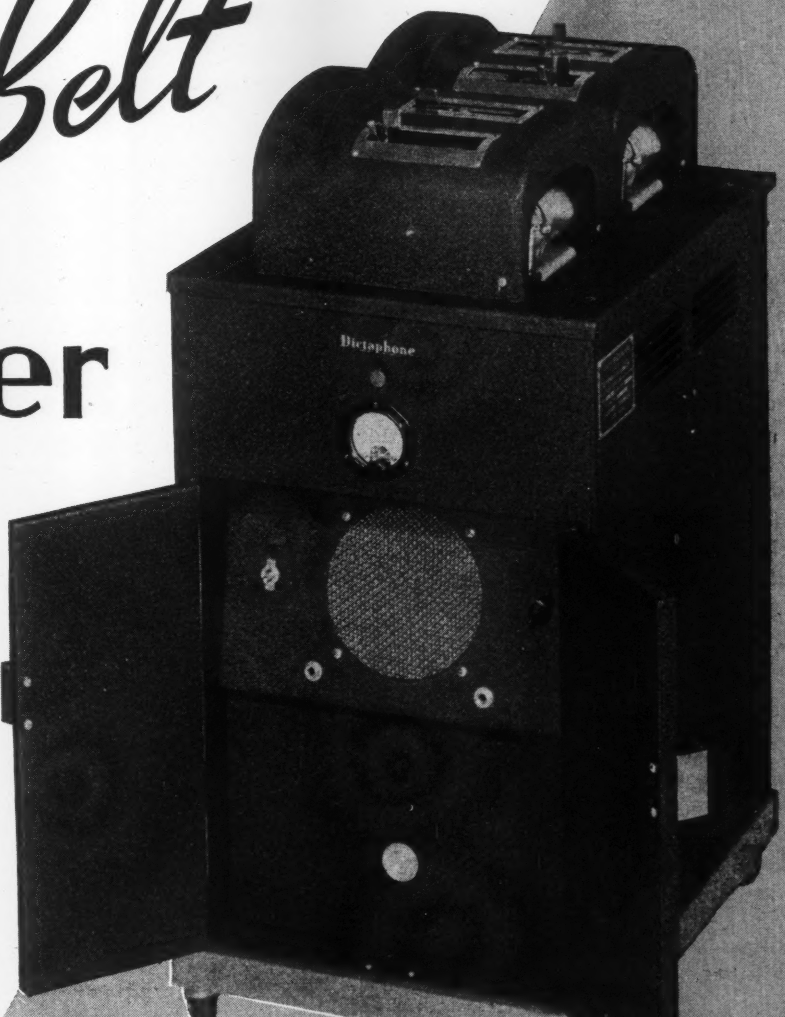


Fig. 1—Overall view of the endless belt recorder with doors open showing speaker

DESIGN specifications of the machine shown in *Fig.* 1 called for a unit providing continuous recording of speech, being adaptable to mass production, and using a low-cost recording medium. The recording medium employed is a collapsible plastic cylinder, or endless belt, 0.005-inch thick, 3½ inches wide, and 12 inches in circumference, into which a tone groove is pressed or "embossed" by a sapphire stylus. The

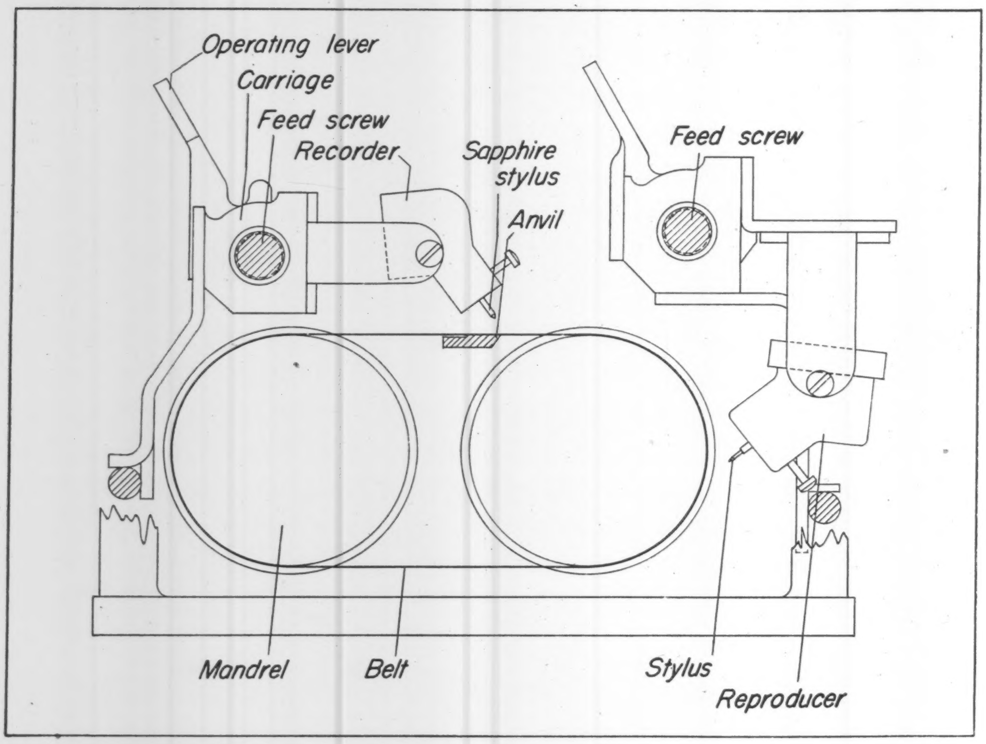

Operating lever
Carriage
Feed screw
Recorder
Sapphire stylus
Anvil
Feed screw
Mandrel
Belt
Stylus
Reproducer

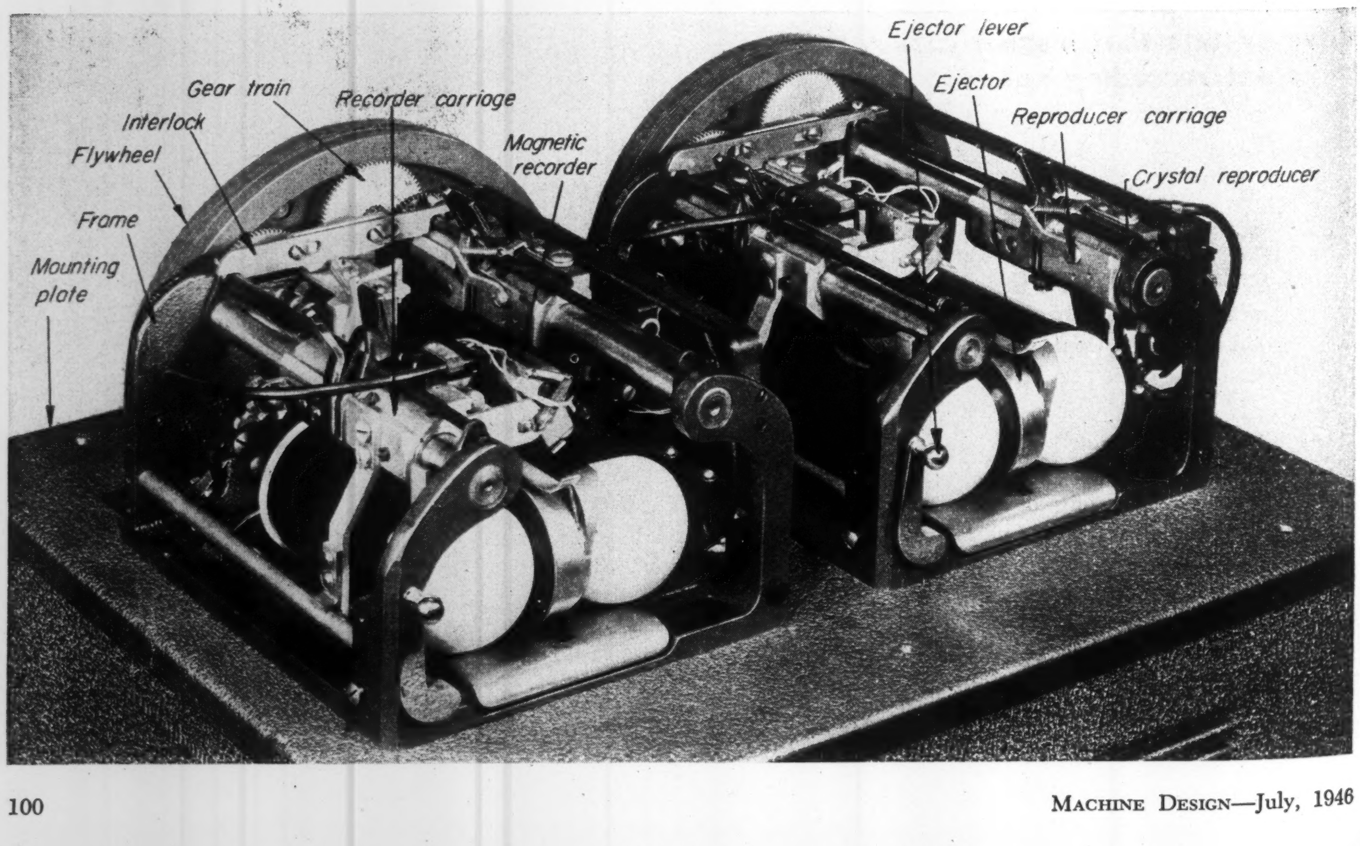
Fig. 2—One of the two recording-reproducing units, showing how the endless plastic belt is stretched between the mandrels. Mandrel at left can be swung in toward drive mandrel, right, to facilitate mounting or demounting of the belt type record

belt may be folded flat for mailing or filing.

In operation, the plastic belt is stretched between two mandrels, *Fig.* 2, one of which is spring loaded. Use of the belt-type record permits constant tone-groove speed throughout the recording interval, while the stretching operation permits the employment of a thinner record than used in previous recording systems. This latter feature means that more sound can be recorded per pound of plastic.

Schematic drawing, *Fig.* 2, shows the drive and driven mandrels with the recording belt in place. Pictured also are the recording and reproducing heads together with their respective feed screws. Recording takes place upon the fixed anvil which is finished in polished hard chromium. Reproduction, either simultaneous with or independent of recording, may be accomplished by the reproducer unit whose stylus contacts the belt as it curves around the drive mandrel. To permit continuous recording, two of the units shown in *Fig.* 2 are assembled for sequential

Fig. 3—Below—The two duplicate recording-reproducing units employed, with key components identified


Gear train
Recorder carriage
Interlock
Flywheel
Magnetic recorder
Frame
Mounting plate
Ejector lever
Ejector
Reproducer carriage
Crystal reproducer

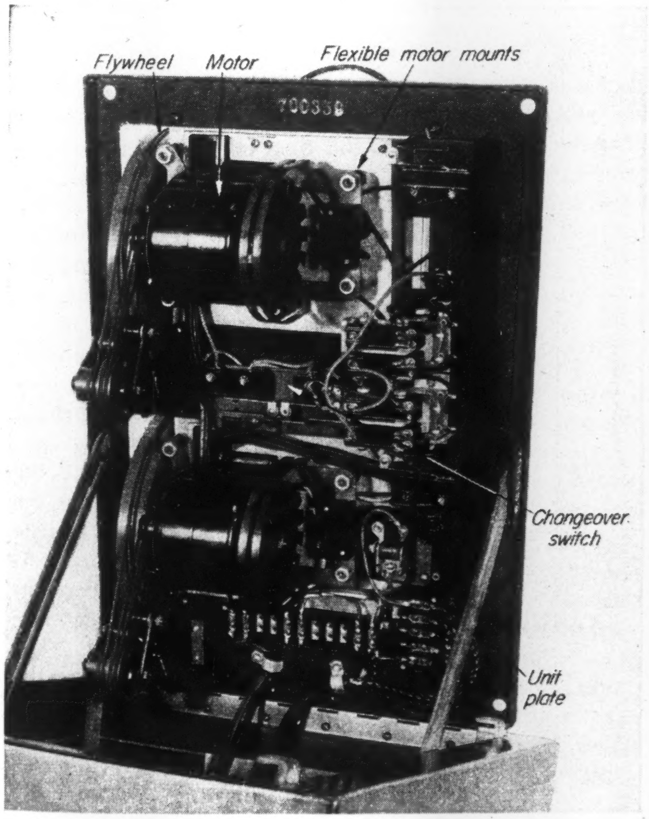
operation. Changeover from one unit to the other with a thirty-second overlap is automatically accomplished. One unit may be reloaded while the other is recording. The recording time per record is one-half hour.

Cabinet of the machine provides mounting space for an amplifier as well as a support for the plate on which are mounted the recording units. This "unit plate" is a 12-gage steel sheet with the edges bent down and the corners brazed. It is provided with cutouts for the frames of the units. These frames, *Fig.* 3, are one-piece aluminum castings, light, yet rigid, designed to be milled easily by ganged cutters, and drilled and tapped in jigs. All projections on the frame are dimensioned to be free from resonance with moving parts of the machine. The bearings used are sintered bronze.

Heart of any recording machine is the drive system which must be free from vibration and rapid speed changes, called flutter. Variations in record speed which have rates of change faster than approximately eight cycles per second must be less than 0.5 per cent. Long-time variations with periods of several seconds up to several minutes may have speed changes of several per cent without noticeable effect on the intelligibility of reproduction.

A drive system using a vibration-free motor and high precision gears can be made to fulfill the above criteria, but such a solution is costly. In the present design this problem was solved through the use of a mechanical filter system in which a shaded-pole induction motor, suspended on springs, drives a flywheel through a double string-belt reduction system, *Figs.* 4 and 5. The belt, of the type used on dental equipment, is used double-wrap to minimize the speed change caused by the belt joint passing over the drive pulleys. The idler guides the cross-over strand and also takes up slack. Pulleys are screw machine parts with pressed-in, oil impregnated, sintered bushings. Idler arms and the mounting bracket are blanked and formed sheet metal parts.

Fig. 4—Below—Unit plate tilted up as for servicing, showing spring-suspended motors and related components



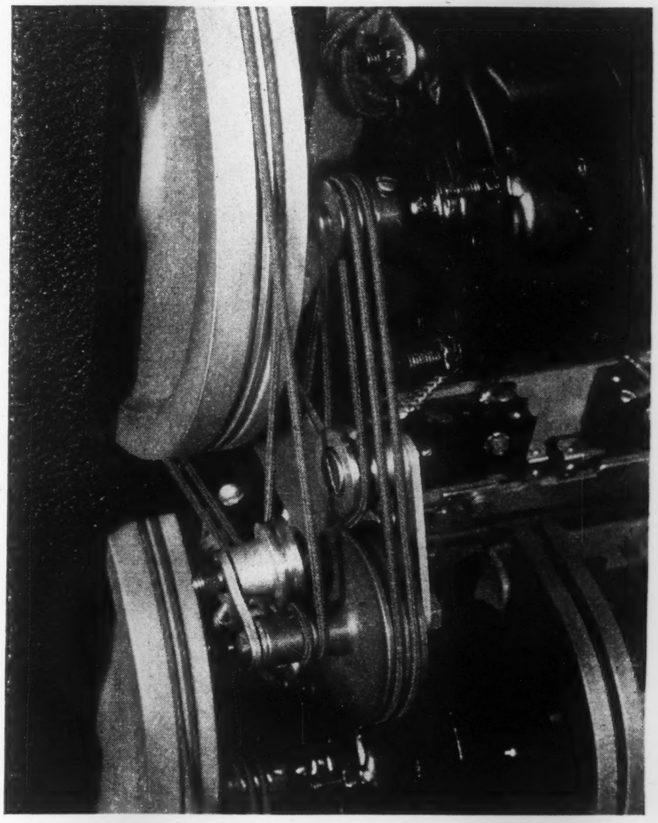


Fig. 5—String-belt reduction system has double wrap to minimize effects of belt joint passing over pulleys

The combination of the mass of the flywheel, the spring in the belt, the idler tension springs, and the friction in the system gives a critically damped mechanical filter. Such a system may be made to have "motion" equal to a fine gear drive at much lower cost and with no lubrication troubles.

Remainder of the drive system consists of a drive mandrel, *Fig.* 6, directly-coupled to the flywheel together with a spring-loaded idler mandrel which maintains the correct position and tension of the recording belt. The idler mandrel is mounted upon an arm, *Fig.* 7, which allows the mandrel to swing in an arc toward the drive mandrel to permit removal of the recording belt. The arm pivots in an adjustable bracket mounted to the main frame casting by means of three screws which are tightly fastened in the frame. Adjustment is made by rotation of the serrated lock nuts shown on both sides of the adjustable bracket . The nuts are adjusted until the axes of the idler and drive mandrels are parallel in the plan view, but slightly out of parallel in the elevation view. This tilt, in a direction such that the rounded end of the mandrel is slightly higher than the flange end, gives the belt a tendency to crawl toward the flange, keeping it on the mandrels during the course of recording. Since the mounting bracket is adjustable, machining tolerances on the individual parts of the assembly are not critical.

Probably the most complex mechanical function of the

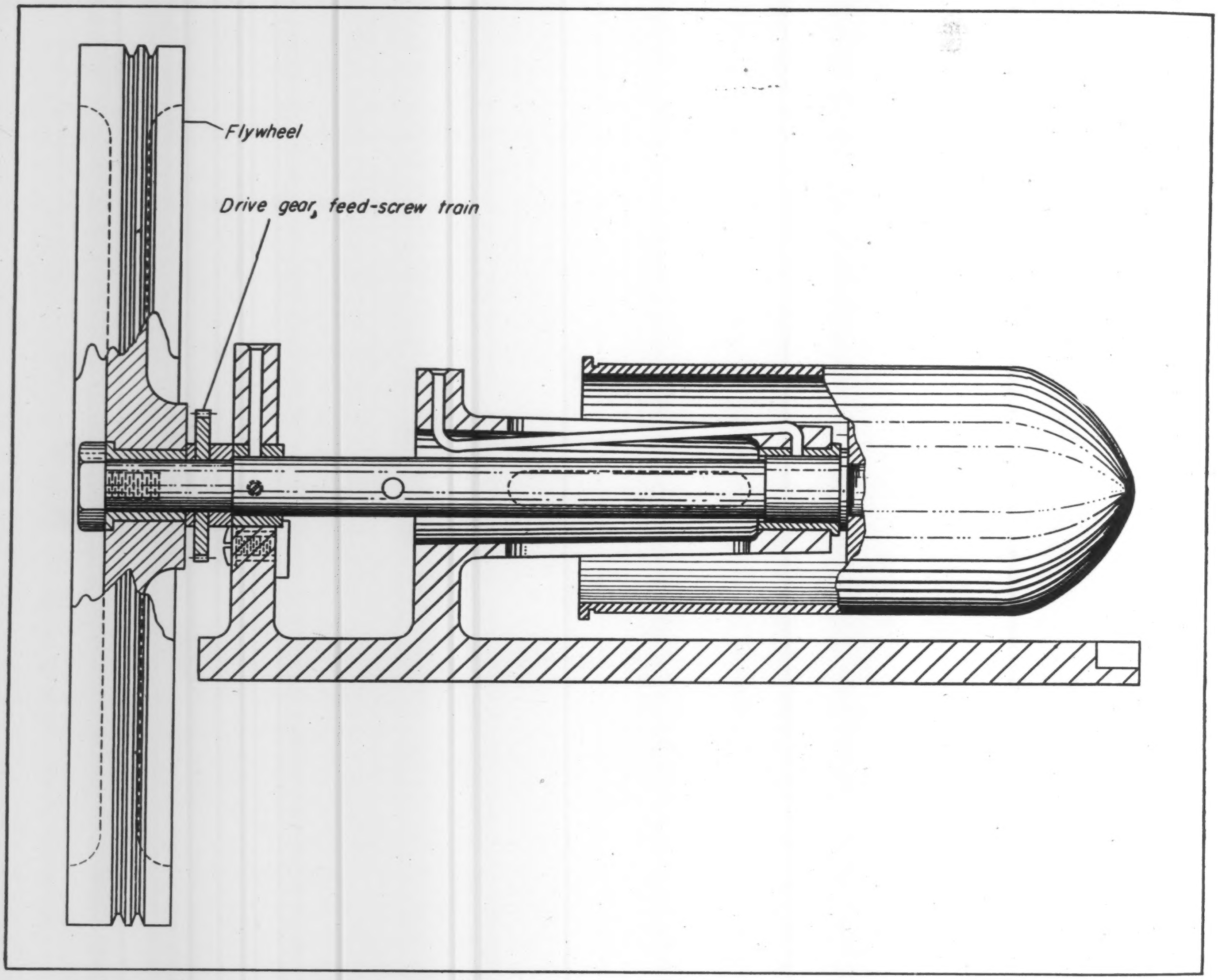
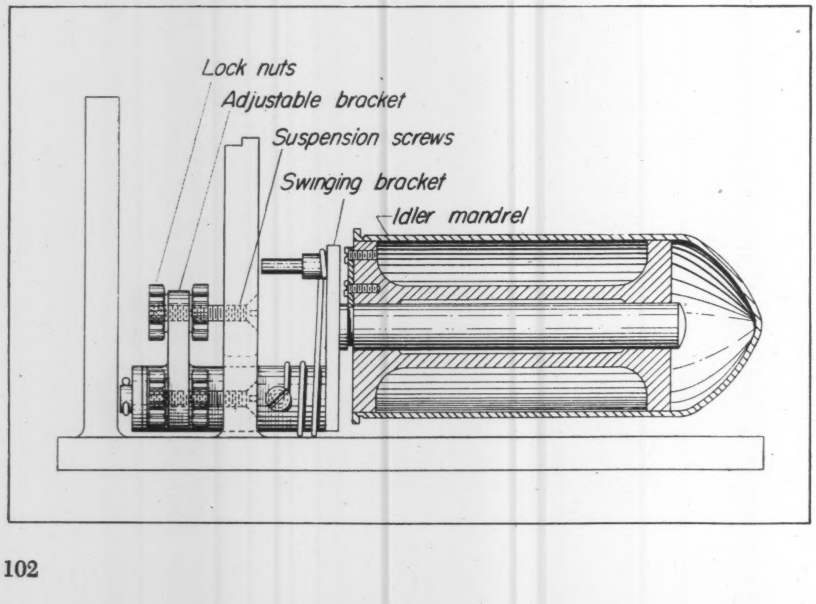


Fig. 6—Above—Construction of the drive mandrel which is direct-coupled to flywheel and carries gear for driving feed-screw gear train

Fig. 7—Below—Idler mandrel is mounted on swinging bracket which can be tilted by adjusting the serrated lock nuts



machine is accomplished by the injector-ejector mechanism, which is located midway between drive and idler mandrels, *Fig.* 3. This mechanism first slides the belt into place, and then releases a latch which allows the spring-loaded idler mandrel to swing against the belt and apply the proper amount of tension. The process is reversed during removal of the belt. This mechanism is pictured in *Fig.* 8 which shows it in both the ejected and injected positions. Inspection of the drawing will reveal two hooks, or fingers, which are retracted in the extended or ejected position of the assembly. The belt is slipped over the assembly while it is in this extended position. Installation of the belt is then completed by pushing against the end of the assembly. As this assembly slides to the left, the hooks rise on the cam to engage the belt and draw it onto the mandrels. At the end of its travel, the injector-ejector latches in place and releases the idler mandrel which sets the belt in tension. The belt is ejected by manually operating the ejector lever, *Fig.* 3 which, swings the idler toward the drive mandrel where it locks in place and then releases the injector-ejector mechanism, allowing the assembly

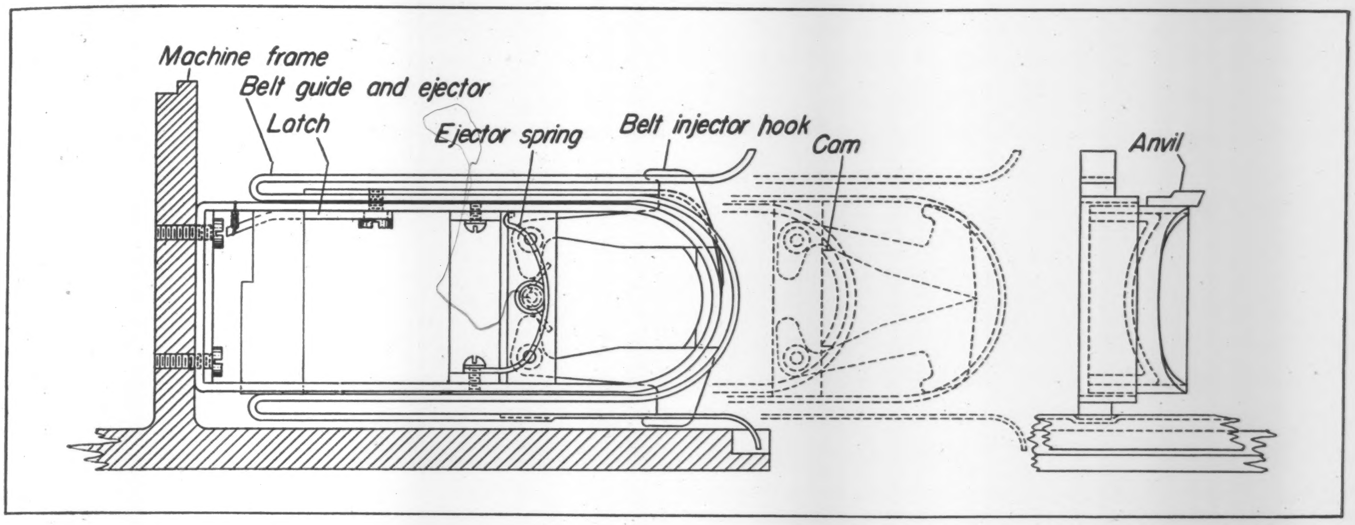
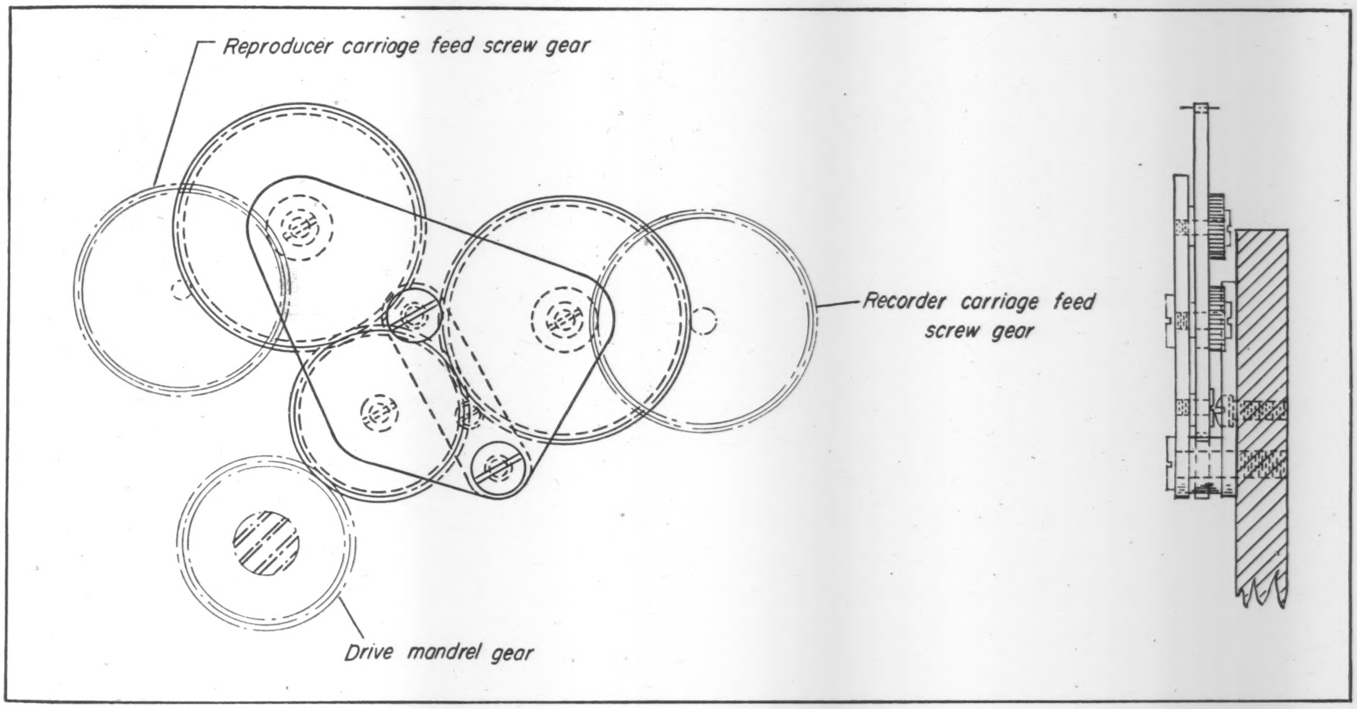


Fig. 8—Above—Injector-ejector mechanism used to slide recording belt into place or remove it. Injector hooks retract as mechanism slides belt off mandrels

Fig. 9—Below—To permit accurate adjustment of mesh, the three intermediate gears between the drive and carriage feed-screw gears are mounted on a separate plate


Reproducer carriage feed screw gear
Recorder carriage feed screw gear
Drive mandrel gear

to snap outward, carrying the belt with it.

To emboss the tone groove on the record in the form of a spiral groove with a pitch of 200 lines per inch, it is necessary to have a positive drive between the drive mandrel and the feed screws which advance the recording and reproducing carriages, *Fig.* 3. The 64-pitch buttress thread feed screws are enclosed in slotted stainless steel sleeves, which also serve as guide rods for the carriages. A spring-loaded feed nut, attached to the carriage, engages the feed screw and transmits thrust to the carriage. To eliminate noise transmission in the feed-screw gear train, use has been made of a laminated phenolic gear meshed with steel driving gears. These gears, for ease of adjustment, have been mounted on a plate, *Fig.* 9, which is in turn mounted on an adjustable bracket fastened to the frame casting. In the design of the carriages, use was made of extruded bar stock for the carriage bodies. Remainder of the carriage parts are fabricated from sheet metal.

The recorder, *Fig.* 10, which must vibrate a sapphire stylus up to 4000 times a second, is a precision instrument built up from sheet-metal parts by the use of precision welding and machining fixtures. This assembly, complete with its permanent magnet which is a cast cylinder with cored mounting hole, is pictured in *Fig.* 11. The only machining operation on the magnet is a grind to length. Parts designated as *A* and *B* are brass, brazed in place. After this operation the entire assembly, less magnet, is annealed.

To obtain a uniform product it is essential that the position and dimension of slots *C* be held to a high degree of accuracy. In order to secure such accuracy, the slotting operation and all drilling and tapping operations are performed in jigs. The entire assembly is plated upon completion of the machining operations and area *D* receives a coil, cemented in place. Completing the unit are the armature, which is coined from soft iron; a blanked beryllium-copper flat spring; and several straps formed from sheet brass.

In addition to the mechanical units discussed in the foregoing, this machine includes a reproducing amplifier, a recording amplifier complete with automatic volume control and a set of remote signal lights.

Fig. 10—Below—Recording head is precision instrument built-up of sheet metal parts. Sapphire stylus is used

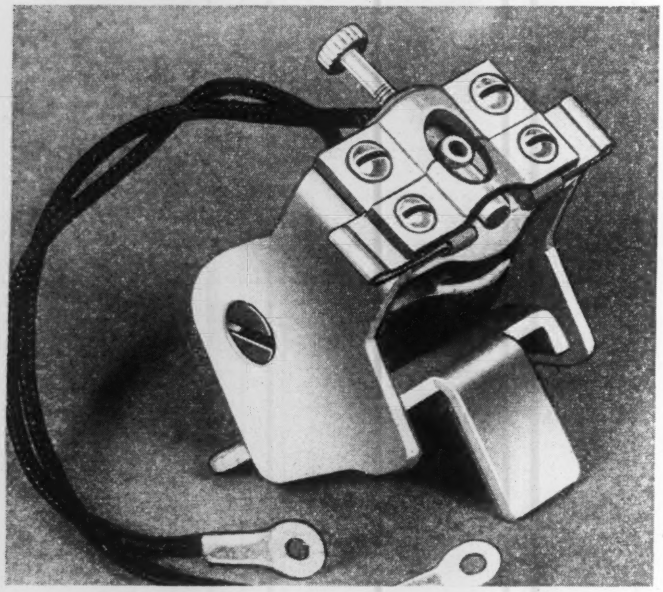
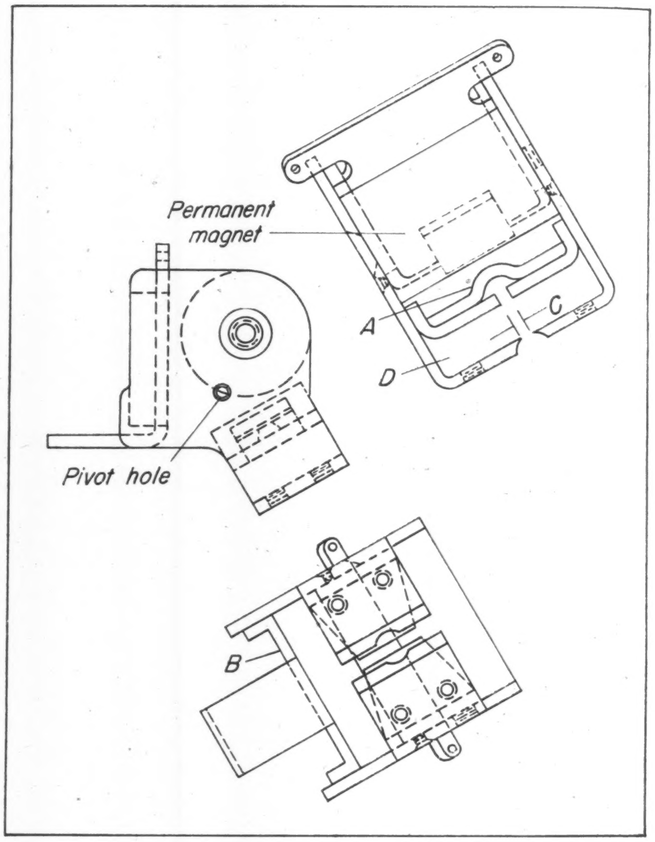


Fig. 11—Below—Assembly drawing of recorder head Parts A and B are brass, brazed in place. Area designated D receives a coil which is cemented in place



How Fury of Bikini Blasts Will Be Measured

When the two atomic bombs are set off near Bikini Atoll, the fury of air and water in the shockwave and blast following the explosions will be measured with intricate equipment capable of recording pressures far greater than those that occurred a half-mile from the Nagasaki explosion. According to the builders of some of the equipment, Westinghouse Research Laboratories, the devices will be placed on unmanned target ships and will transmit automatically by frequency-modulation radio their information to receivers on mother ships.

In the first test, recordings will be made of the air pressure exerted on two target ships by explosion of an atomic bomb dropped from the air and timed to explode several hundred feet above the surface. The second test will be made during the explosion of the second atomic bomb on the water's surface and will measure the pressure created at six positions around a target ship.

Complicating the transmission and recording of the water pressure data is the fact that great fluctuations in intensity are anticipated, and equipment had to be devised to record significant changes that are expected to occur in as little as one fifty-thousandth of a second.

In the air pressure test, similarly significant changes probably will occur much more slowly—or in approximately one two-hundredth of a second. Thirty-six gages will transmit air pressure data from 18 positions on each of two target ships over radio channels. Electrical impulses from each of the 18 gages on a radio channel will be examined by an electronic commutator every few millionths of a second and transmitted successively from each gage to two receivers on the mother ship. A brass bellows in each gage will stretch a resistance strain gage when compressed by the blast of air and this strain gage in turn will frequency-modulate a radio transmitter by means of a reactance tube.

For the second test, six tiny tournaline crystals sealed at the ends of rubber cables will send electrical impulses over six separate radio transmitters in a target ship. The cables themselves will run through the ship's hull into the sea. When the water pressure from the bomb squeezes the crystals, they will produce electrical impulses.

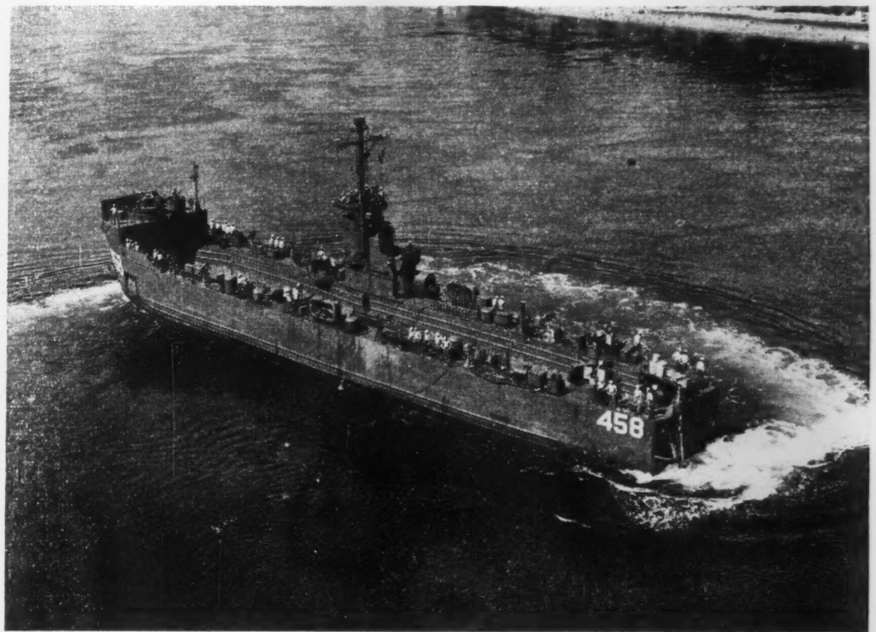
Because sound travels three to four times as fast in water as it does in air, the impulses will be correspondingly more rapid and transmitters and receivers must be exceedingly precise. Three cathode-ray tubes—an advanced model of those used in television receiver sets—will pick up and amplify the "picture" made on their fluorescent screens by the impulses. To photograph these curves, two high-speed cameras are installed in the recorder. Changes that occur in a fifty-thousandth of a second are photographed by the cameras, which use up film at the rate of 40 feet a second.

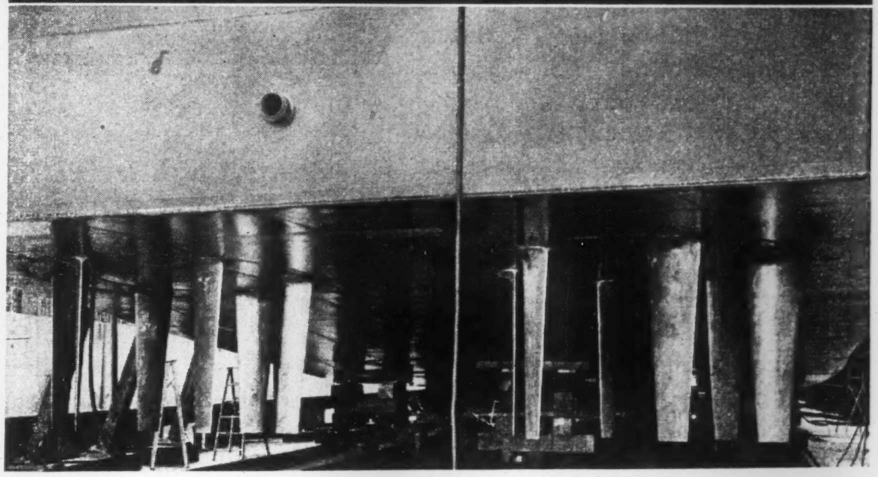
Scanning THE FIELD for Ideas

CYCLOIDAL propellers installed on the Navy LMS 458, shown below give this ship extreme maneuverability without regard to its speed. Underwater installation of these propellers, originally designed by Professor Frederick K. Kirsten in 1921, are shown in the photograph at right. In Austria, this form of propulsion was used as early as 1928, but it was not till 1945 that the U. S. Army and Navy decided to equip a ship in this manner.

Having an orbit diameter of 7 feet, each propeller is equipped with six hollow-cast blades of manganese bronze. A rotor which carries the blades is driven through a spiral bevel gear. Direct-connected to the engine shaft by a flexible coupling, the pinion shaft consists of an inner pinion integral with the shaft and an outer pinion keyed to the shaft. These two pinions drive two spiral bevel ring gears mounted on a common spider which in turn is keyed to and drives the rotor shaft. The gear system is so designed that the pinion shaft is mounted on roller bearings and is free to float endwise. Spiral angle of the gears is cut opposite and of the proper relation to proportion the gear loads correctly and balance the pinion thrust.

Blade mechanism has a central control shaft which mounts the sun gear in the rotor and is normally stationary except when used for steering. Three idler gears mesh with the sun gear, and each idler meshes with two blade gears. As the rotor turns, the blades revolve around their own axes at one-half the rotor speed, the ratio between the blade gears and the sun gear being 2:1. Rotative motion of the blades is controlled so that each blade produces a useful thrust during practically its entire revolution on its orbit. In this unusual system of ship propulsion all





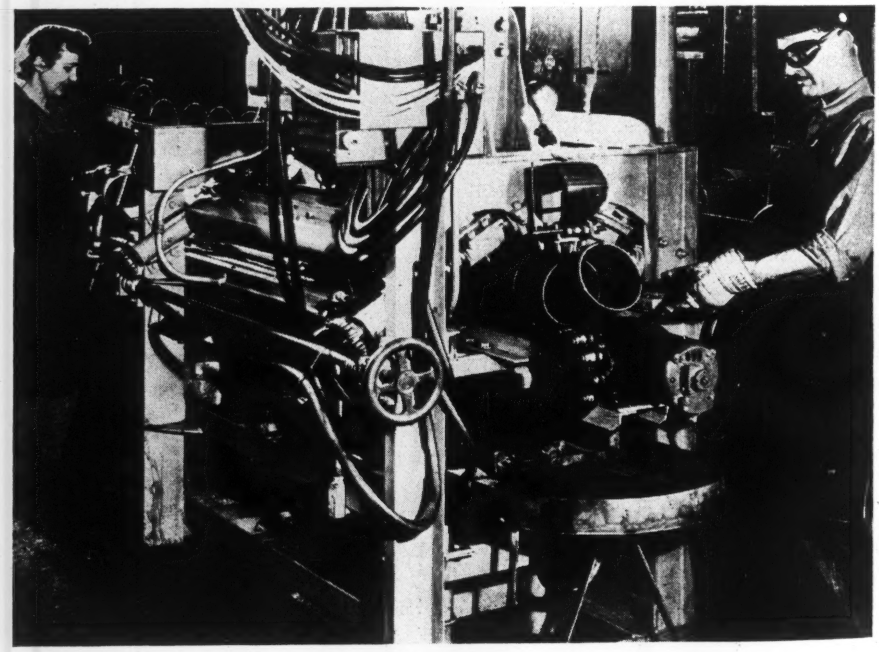
bearings and gears in the transmission are force feed lubricated. The oil is circulated through strainers, heater and coolers by external shaft driven pumps with electrical standby units. Double blade seals are constantly under oil pressure even when the propeller is stationary. A constant head is maintained in excess of the external water pressure to insure no leakage of water into the propeller. The main shaft seal consists of three parts. Water is circulated between the two lower units for lubrication, and oil pressure is maintained between the two upper units.

Steering is accomplished in the propeller by the control shaft which, when turned by a worm and worm gear on the top, shifts the axis of symmetry of the propeller. In this way, all blades may direct their thrust to effect steering. The ship does not depend upon motion for maneuvering whereas, a screw driven ship, depends upon sufficient speed for rudder effectiveness. Propeller controls are fully pneumatic and are combined with pneumatic engine controls in the same stand, shown above. This control is located on the bridge. For emergency operation there is a duplicate stand in the propeller room.

To compare the efficiency of the cycloidal propeller with a screw propeller is difficult. Under certain conditions, however, the cycloidal unit is superior to the screw. Vibration problems inherent in screw propeller designs are not present because the blades of the cycloidal propeller do not approach and recede from the hull.

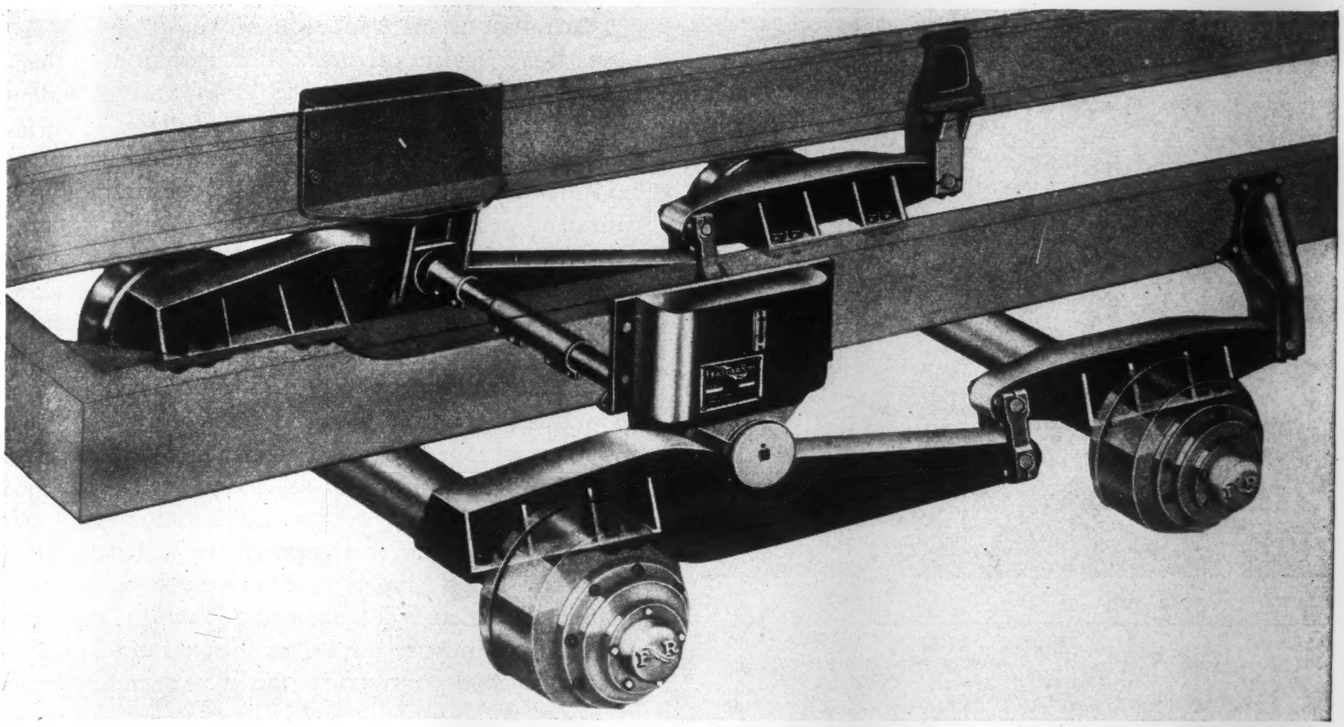
In seam welding of rounds, squares and other shapes of tubes and tanks it has been customary to clamp the work in a fixture and move the welding head across the seam. To obviate operations such as breaking the weld, unclamping the finished work and clamping the new piece, the welding machine shown below employs a stationary head with the work moving under the head in a continuous stream. Designed by the Victory Engineering & Machine Works Inc., the machine requires only an operator to place the pieces on the feed end and another to remove the finished pieces at the discharge end. Production runs average 60 to 150 inches per minute.

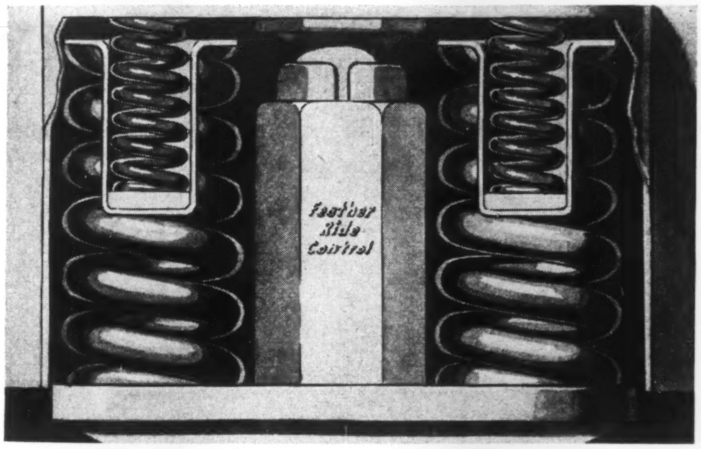
Utilizing a chain type drive, practically a continuous line contact is made with the work. This allows short lengths to be run consecutively, as well as long ones. Gripping of the line contact is firm but does not leave marks of the fluted drive rolls on the workpiece.



Sturdy, lightweight suspension for tandem axles on highway trailers, increasing load carrying capacities while at the same time reducing the weight and complexity of the running gear, is shown at top of next page. Developed by R. W. Pointer of Feather Ride Corp., the suspension utilizes novel linkages, coil springs and snubbers to provide load carrying capacities up to 200,000 pounds. Features include underload springs, elimination of axle kick-up on fast stops, no need for lubrication, automatic axle-trailing to reduce tire scuffing on curves, and a load indicator.

Nested in the high-load carrying springs are cups containing inter-





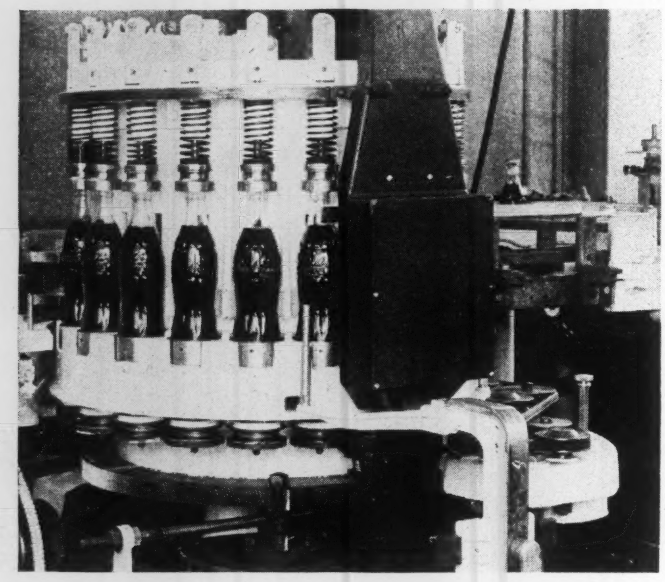
mediate springs so that when the trailer is traveling light or unloaded it will have a cushioned ride. This unit is shown in the above cutaway view. Between the main springs is a snubbing device to interrupt the rhythm of the springs. Three metal shoes in the snubber slide up and down inside a tapered cylinder. Upon impact from road shock these shoes expand against the sides of the tapered cylinder by means of a wedge action. After thus resisting motion, they return to normal position by an inner coil spring and are ready to act again. This frequency snubber is self-lubricating, requires no attention, and is designed for a half million miles of service before requiring replacement.

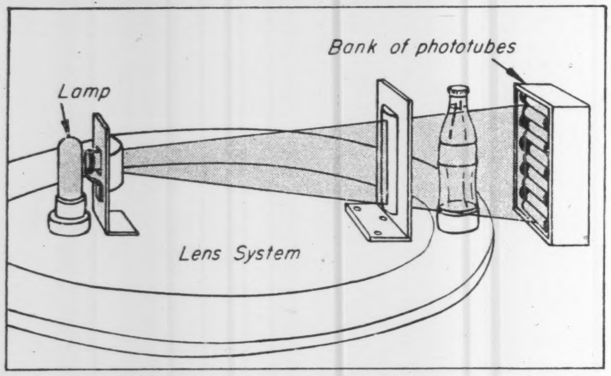
Designed for small deflections the system affords stability, minimum side sway and exceptional control of the vehicle. Travel allowance of the main coil spring is only two inches. At rated capacity of 17,000 pounds per axle the springs are designed to deflect only 40 per cent or 0.8-inch. As illustrated, the load is suspended in front of the axle center. Upon application of the brakes each axle twists the linkage in a rotary motion, exerting a definite effort to pull downward at point of attachment to the frame. The greater the speed or load, the greater is the downward pull exerted. Completely rubber bushed, these mountings allow the axles to give sideways under the frame, conforming to the radius of curves.

Suspension is hundreds of pounds lighter than conventional designs. Main coil spring itself weighs only 15 pounds. Elimination of leaf-spring accessories like multiple shackles, shackle pins, spring clips, tie-bolts, U-bolts, radius rods, torque arms, compression bars, etc., also contribute to saving of weight and increasing of payload.

Automatic inspection of bottled beverages has long been a major concern of the beverage industry. Foolproof inspection, however, is far from a simple task even with the aid of phototubes. Electronic, optical and mechanical engineering features must be skillfully coordinated to provide a dependable design.

Shown at top of next page is an inspection machine developed by RCA for the Coca-Cola company which has been proved by four years of field testing in bottling plants. Lamp and lens system sends a strong, narrow beam of light through the length of the bottle into a bank of phototubes. In the photograph the narrow aperture for admitting the light to the bottles may be seen behind each bottle on the



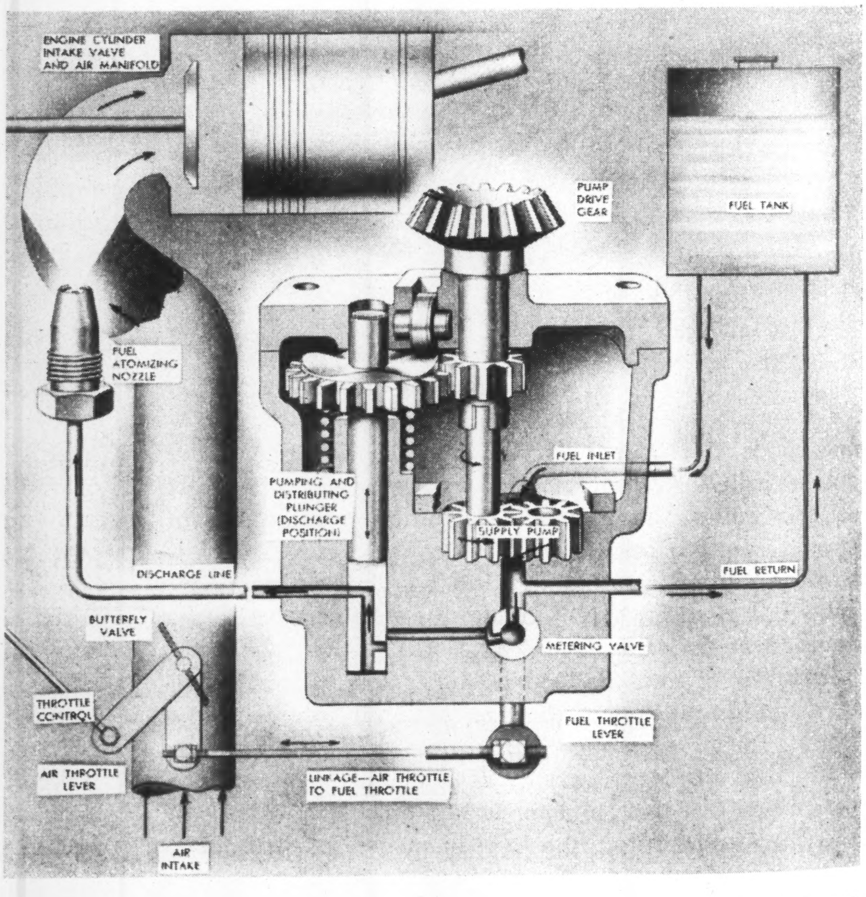

Bank of phototubes
Lamp
Lens System

turntable. In the foreground to the right is seen the housing enclosing the bank of phototubes. A schematic diagram of the electronic inspection system for this machine is illustrated above.

These tubes and the associated electronic amplifier are so designed that a minute change in transmitted light falling on the tubes results in a strong electric current to the selector mechanism. To detect a foreign particle in the beverage the particle must be made to move through the light beam. To accomplish this, the bottle is first spun at a high rate so that the contents spin with the bottle. Then the bottle is quickly braked, leaving the contents to whirl during the inspection period. Lettering and imperfections in the glass bottle do not cause changes in the light and therefore have no effect on the operation of the selector mechanism.

Fuel injection systems on small aircraft engines have design advantages in addition to eliminating danger of carburetor icing and providing even distribution of fuel to all cylinders. Flat injector engines allow for better concealed wing and hull installations, obviating duct work normally required to provide carburetor heat.

Novel features of a compact injection pump are shown schematically in the system diagram below. Designed by Ex-Cell-O Corp., the pump consists of a supply pump, a fuel-metering valve, and a pumping and distributing plunger. Driven through gears at one-half crankshaft speed on a 4-cycle engine, the plunger operates in a steel sleeve and is supported on a roller which rests against a cam face on the upper side of the gear. The number of lobes or lifts on the cam corresponds to the number of cylinders; these give a plunger stroke of approximately 0.05-inch. The metering valve is linked to the air throttle lever on the air-intake manifold. Both are calibrated to provide the exact fuel-air ratio required and both operate in response to the throttle-control lever. Fuel enters the supply pump and is metered to the plunger through inlet ports that come into registry with the feed duct on the suction stroke. On the downward or discharge stroke, a discharge port in the plunger registers with an outlet port in the sleeve and the fuel passes on under pressure through a discharge line to an atomizing nozzle.


ENGINE CYLINDER INTAKE VALVE AND AIR MANIFOLD
PUMP DRIVE GEAR
FUEL TANK
FUEL ATOMIZING NOZZLE
FUEL INLET
PUMPING AND DISTRIBUTING PLUNGER (DISCHARGE POSITION)
SUPPLY PUMP
DISCHARGE LINE
FUEL RETURN
BUTTERFLY VALVE
METERING VALVE
THROTTLE CONTROL
FUEL THROTTLE LEVER
AIR THROTTLE LEVER
LINKAGE—AIR THROTTLE TO FUEL THROTTLE
AIR INTAKE

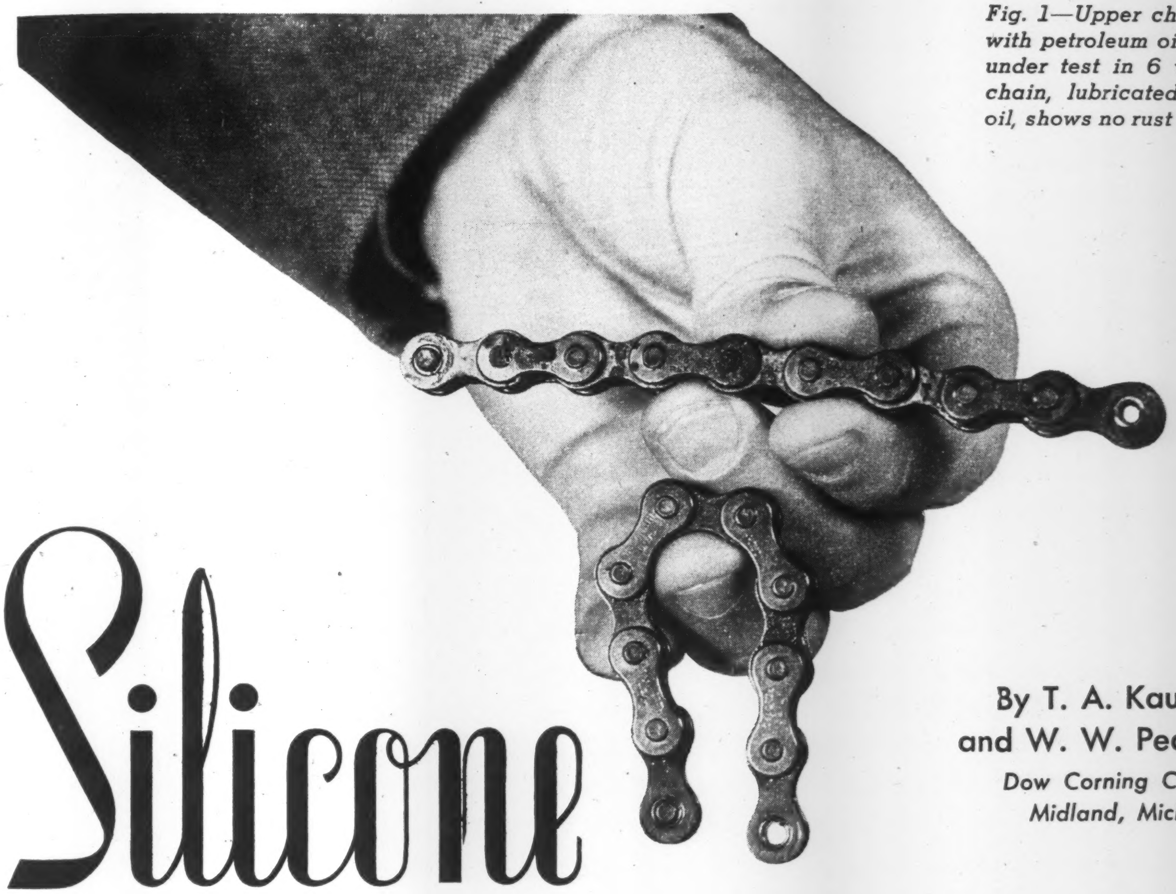


Fig. 1—Upper chain, lubricated with petroleum oil, rusted badly under test in 6 weeks. Lower chain, lubricated with silicone oil, shows no rust after 21 weeks

Silicone Oils and Greases

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Dow Corning Corp.
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Silicone fluids offer possible solutions to lubrication problems involving heat stability, oxidation resistance, nonvolatility and low change of viscosity with change in temperature. One silicone grease promises complete stability for permanent lubrication of ball bearings

SILICONE lubricants possess combinations of properties which go a long way toward eliminating problems which for many years have plagued the lubrication engineer. For example, thinning at high temperatures and thickening at low temperatures—effects customarily encountered with conventional lubricants—are greatly minimized in silicone fluids. The viscosity-temperature slopes of a family of silicone fluids are compared in *Fig.* 2 with some typical petroleum oils. As will be seen, if a silicone fluid and a petroleum oil of high "viscosity index" both have a viscosity of 50 centistokes at 100 F, then at zero F the petroleum oil will have six times the viscosity of the silicone fluid.

Many lubrication problems involve a rather delicate balance between obtaining an oil with low enough viscosity and one with sufficiently low volatility. For example, a kerosene with no tendency to evaporate would be highly desirable in some applications. While silicone fluids do not quite meet this specification, it is possible to obtain nonvolatile silicone fluids in the viscosity range of 3 to 10 centistokes.

Low temperatures such as are encountered in aircraft applications often force a compromise between pour point and volatility or between pour point and viscosity. That is, a petroleum oil of low enough volatility and high enough viscosity for a given application may have much too high a pour point. The Dow Corning 200 silicone fluids,

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From papers presented before the Fuels and Lubricants section of the Society of Automotive Engineers at the March 15 meeting of the American Society of Lubrication Engineers in New York.

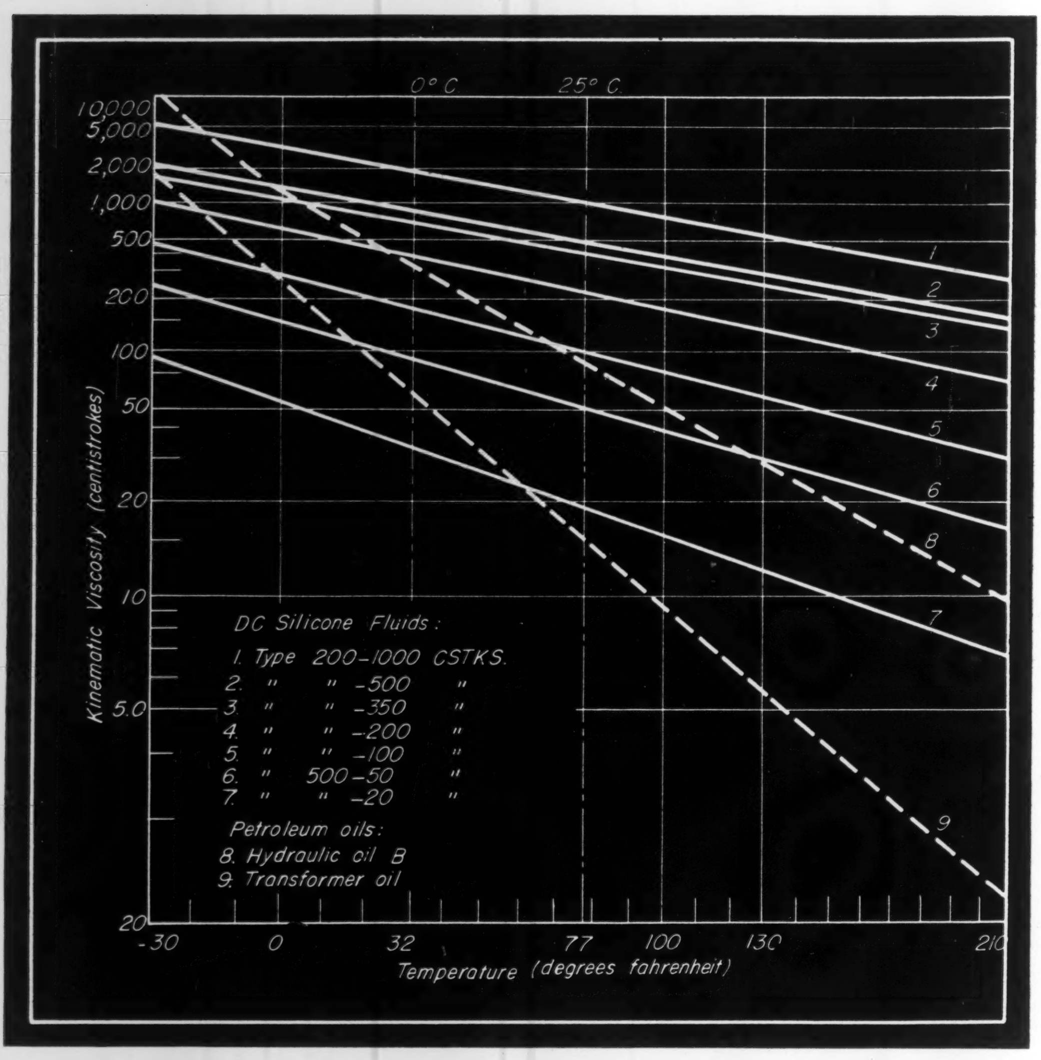


Fig. 2—Viscosity-temperature slopes of two types of silicone fluids and two types of petroleum oils graphically show the superior stability of silicone fluids

which are nonvolatile and can be obtained at viscosities of 1000 centistokes and higher, all have pour points below −50 F.

Utility of properties such as heat stability, nonvolatility and oxidation resistance is illustrated in *Fig.* 1. The chain pictured was subjected alternately to high temperature (350 F) and high humidity in a carefully controlled test setup. After six weeks of operation, the length of chain lubricated with petroleum oil was rusted so badly that the rollers would no longer turn, largely because the high temperature condition had oxidized and evaporated the petroleum oil. The chain treated with silicone fluid (DC 710 R) was still in excellent condition after 21 weeks. It showed very little rusting and still had a protective film of the silicone lubricant.

LUBRICATING PROPERTIES OF SILICONE FLUIDS: One of the first experiments conducted to determine the lubricating properties of silicone fluids involved pumping the

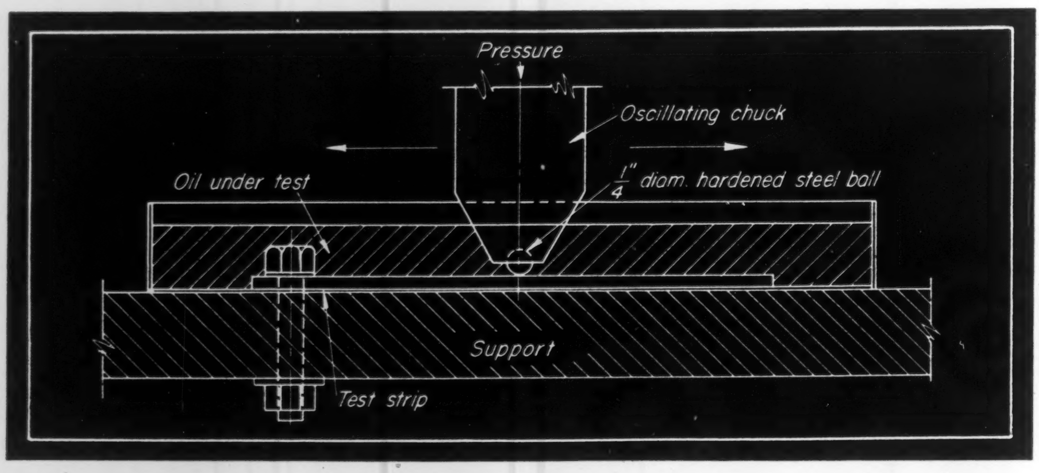


Fig. 3—Diagrammatic drawing of lubricant testing device for measuring wear between sliding surfaces over a range of low operating pressures

fluids with a small gear pump at various outlet pressures. The pump was operated at approximately normal rated speed. An initial test on an SAE 10 motor oil showed that no appreciable wear occurred during 10 days of operation at outlet pressures ranging up to 100 psi. A run made with DC 200 silicone fluid of comparable viscosity produced extreme wear, as indicated by suspended particles of steel in the fluid, within one hour of operation at 25 psi outlet pressure. The wear had occurred mostly in the steel bushings supporting the gears and between the gears and the steel sideplates; apparently very little wear had occurred between the gears themselves.

From this and similar experiences with other types of equipment it became apparent that the DC 200 fluids do not have the lubricating properties of an ordinary petroleum oil. They have no extreme-pressure lubricating properties and actually begin to fail at unit pressures considerably lower than those normally accepted as the beginning of the extreme pressure range.

TABLE I

Wear Tests on Metal Combinations
2-hour test, 1500-gram load
(wear expressed in 0.0001-gram)

Hardened Steel Ball Against	Lubricant SAE 20	DC 200	DC 550	DC 710
Brass	4	130	40	3
Aluminum	65	250	430	300
Magnesium	10	40	45	4
Soft Steel	0	750	330	0
Hardened Spring Steel	2	4	3	3

WEAR TESTS: In conducting tests of wear between variously lubricated sliding surfaces over a range of rather low pressures, the unit shown diagrammatically in *Fig.* 3, was used. It merely slides a ¼-inch metal ball over a flat metal strip. The ball is held rigidly in a chuck which can be loaded vertically in the range of 500 to 4000 grams and has a horizontal reciprocating motion. Amplitude of the oscillation is 2⅜ inches, and the speed is 72 complete oscillations per minute. The flat metal strip is held stationary in a pan which contains the oil to be tested and the usual method of evaluation is to weigh the test strip before and after a 2-hour run. The loss in weight of the test strip, or in some cases the depth of the groove worn in the strip, is the measure of lubrication efficiency. Results obtained with a hardened steel ball in the chuck and a strip of shim brass as the wearing surface are shown in *Fig.* 4. After two hours the weight loss of the brass strip was plotted against the load applied to the steel ball. It is apparent that the wear which occurred with DC 200 silicone fluid is greater than with the SAE 10 motor oil. Similarly, *Fig.* 5 shows the effect of running time of the tester on weight loss of the brass strip.

TABLE II

Percentage of Weight Lost
After 40 Hours Exposure at 175 C (347 F)

High-Temperature Petroleum Greases			Dow Corning Silicone Greases		
A	B	C	DC 33	DC 41	DC 44
58.3	39.3	34.8	4.6	4.2	3.3

Note: After 40 hours exposure at 175 C, a crust had formed over the surface of the three petroleum greases. The silicone greases were still soft and serviceable.

Subsequent tests showed that variations in the chemical composition of the silicone fluids had a great effect on lubricating properties, and several new types of fluids were developed. Wear tests results on one of the newer silicone fluids are also shown in *Figs.* 4 and 5. In *Fig.* 4, DC 710 shows less wear up to 2000 grams load than SAE 10 petroleum oil, but about twice as much wear at 3000 grams load. As is shown in *Fig.* 5, at 1500 grams load, DC 710 produces considerably less wear than the petroleum oil at all running times up to 16 hours. The tentative conclusion to be drawn from these results is the DC 710 is better than petroleum oil for steel-to-brass lubrication at moderate loads, but is less effective than petroleum oil at loads approaching the extreme-pressure range.

WEAR ON VARIOUS METALS: Results obtained with a hardened steel ball sliding on various metal surfaces are shown in TABLE I. It is apparent that aluminum has the greatest tendency to wear under these conditions, and spring steel the least. DC 710 is a definite improvement over DC 200, and appears equivalent to SAE 20 except in the case of aluminum. It should be noted that test values below 5 are all equivalent, since this is the order of accuracy of the test procedure.

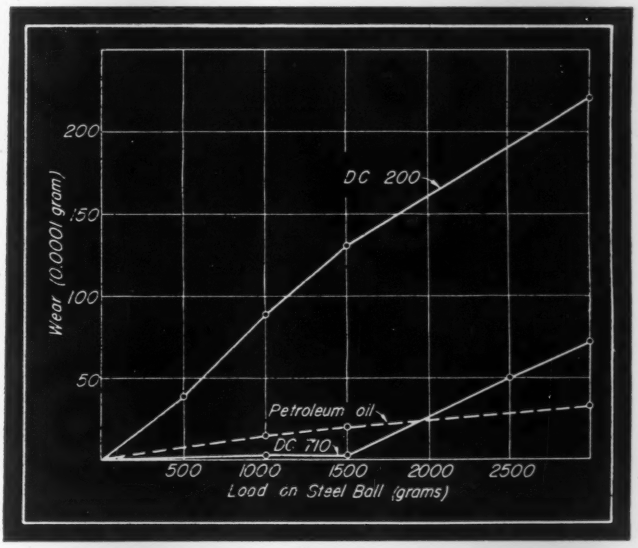


Fig. 4—Wear in terms of weight lost in two hours by a brass strip against which a hardened steel ball, lubricated with SAE 10 petroleum oil and two silicone fluids, is rubbed at a constant rate under various loads

It is interesting to note the relative behavior of DC 200 with soft steel and spring steel, and to compare this with results obtained on the pumping test. It will be recalled that the gear pump cited previously had soft steel sleeve bearings supporting the gears and soft steel side plates and, while excessive wear occurred on these parts, almost no wear was found on the gears themselves. Another test, using a rotary vane-type pump, has been operating with DC 200 with excellent results. In this case, the steel parts are hardened, and the side plates are bronze. The pump has been running for over 2000 hours at 1000 psi pressure, and with the silicone fluid held at a temperature of 90 C. To date, wear has been negligible.

It will be seen that considerable differences in lubricating ability exist between the various silicone fluids and that at least some of them are suitable for a range of lubrication applications. Even the DC 710 fluid, however, is not an extreme-pressure lubricant.

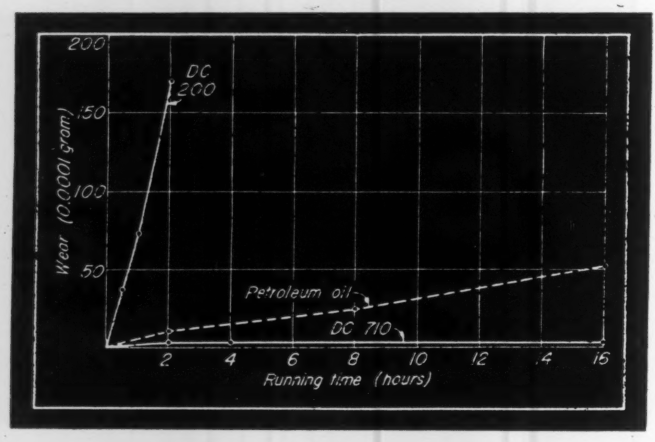
SILICONE LUBRICATING GREASE: Most discussions of the properties required in an oil for making a lubricant of grease-like consistency emphasize exactly those properties for which the silicone fluids are outstanding. Some of these properties are: (a) Low change of viscosity with temperature, (b) low volatility, (c) resistance to oxidation, (d) stability at high temperatures, and (e) low freezing point. When all of the above properties are available in a single fluid, it becomes unnecessary to make compromises in the properties of the final grease.

TABLE III

Specifications of Some Silicone Greases

Property and Specification Test	Type of Grease		
	DC 33	DC 41	DC 44
Penetrometer (worked grease) (ASTM No. D 217-44T)	280–330	295–305	240–260
Flash Point (ASTM No. D 92-33)	525 F	535 F	550 F
Dropping Point (ASTM No. D 566-42) not less than	390 F	390 F	390 F
Bleeding (ANG-3A Specification) 16 hrs at 110 C, less than	2%	4%	2%

Fig. 5—Below—Wear in terms of weight lost by a brass strip against which a hardened steel ball, lubricated with SAE 10 petroleum oil and two silicone fluids, is rubbed at a constant rate and under a constant 1500-gram load for various periods of time



The problem of making a good silicone grease for bearing lubrication was complicated by two factors. First, any thickening agent used should have about the same order of heat stability as the silicone fluids. Second, most of the metallic soaps proved to be insoluble in silicone fluids. However, the problem was solved finally by two different methods and at the present time two types of silicone lubricating greases are being made. One, DC 41, is designed for bearings operating at temperatures up to 200 C and at speeds which are relatively low. The other is produced in two formulations, DC 33 and DC 44. DC 33 is designed for high-speed bearings operating at very low temperatures but it also has good lubricating life at elevated temperatures. DC 44 is for bearings operating at elevated temperatures and at speeds up to 10,000 rpm.

An indication of the low volatility of these silicone greases is afforded by the data listed in TABLE II. Their specifications are given in TABLE III.

SILICONE GREASE DC 41: In this grease, a finely divided carbon black is used as the thickening agent for a silicone fluid of good lubricity. In one series of tests, six No. 212 ball bearings were mounted on a shaft in an enclosure which could be heated to any desired temperature (see *Fig.* 6). Each bearing was spring-loaded with 150-lb radial load and the shaft rotated at 1750 rpm by a 1 hp motor. Failure of the bearings was determined by:

1. Decrease in the time required for the rotating shaft to coast to a stop when the motor was shut off
2. Observation of any abnormal temperature rise of an individual bearing by means of a thermocouple placed in each bearing housing
3. Observation of any sudden increase in power required to drive the test set as indicated on a watt meter.

This test set was first used to evaluate the life of DC 41 grease at 150 C (302 F) and an average bearing life of about 1300 hours was established. At 125 C (257 F) bearing life was 6000 hours. These figures are about eight times the values that were observed with the best petroleum greases.

DC 41 will function as a grease at temperatures of 200 C and above, and in some specialized applications, such

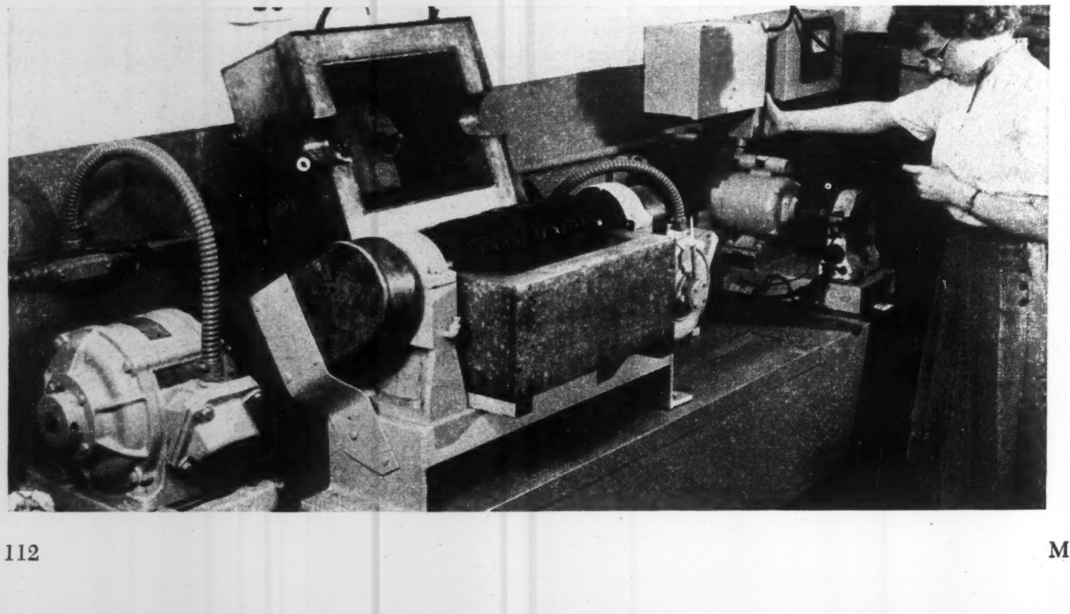


Fig. 6—Left—In test device shown at center, six ball bearings are mounted on a shaft and spring-loaded radially. Shaft and bearings are enclosed in oven which can be heated electrically as desired. Technician is checking coasting time on single-bearing tester

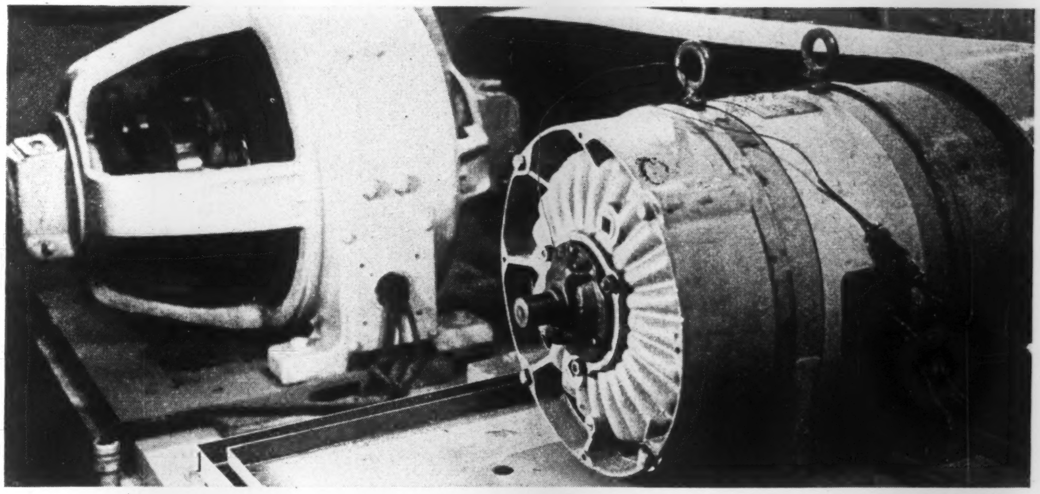


Fig. 7 — Right — Silicone-insulated induction motor is belt-connected to generator and loaded so that bearings operate at temperatures up to 150 C in testing various silicone greases

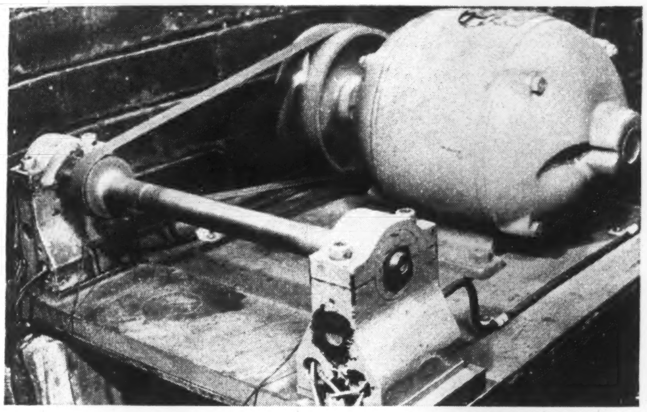


Fig. 8—Grease tester consists of two standard No. 204 bearings mounted in electrically heated cast-iron blocks. Bearings support shaft which is belt-driven at 10,000 rpm. Changes in torque are measured by temperature rise of bearings and changes in hot and cold coasting time are used as indications of bearing failure

as in the pumping of very hot liquids or in bearings which must be located in high-temperature ovens, DC 41 has definite value. Bearings with bronze retainers will give several times the life obtainable with ordinary bearings.

Silicone Grease DC 44: This is a silicone fluid thickened with heat stable metallic soaps. It has been tested quite extensively at both 1750 and 10,000 rpm and at bearing temperatures between 115 C (239 F) and 200 C (392 F). Three different types of tests were made, namely: (1) Single-bearing 1750-rpm tester, (2) bearings in 1750 rpm electric motors and (3) 10,000-rpm tester.

Single-bearing 1750-RPM Tester: A No. 212 bearing with a bronze retainer ring was operated in an electrically heated cast iron housing (see *Fig.* 6). Grease failure was determined by measuring both hot and cold coasting times and by observation of the appearance of the grease at intervals. As is shown in Table IV grease failure occurred after 2600 hours at 150 C, or after 450 hours at 200 C.

Bearings in 1750-RPM Electric Motors: Bearing temperatures were produced by actually loading the motor to the desired extent with a belt-driven generator. This test set-up is shown in *Fig.* 7. Column II in Table IV shows that the two tests, at 115 C and 125 C were still in operation after 3500 and 2500 hours. Motor coasting times, which were taken with the bearings both hot and cold, and the bearing temperature rise, obtained with the motor operating under no load, indicated that little if any change occurred in the bearings or in the DC 44 grease.

Bearings in 10,000-RPM Tester: Shown in *Fig.* 8, this tester consists of two standard No. 204 bearings mounted in electrically heated cast iron blocks. The bearings support a shaft which is belt-driven at 10,000 rpm. Changes in torque are measured by temperature rise of the bearings and changes in hot or cold coasting time are used as indications of bearing failure. As shown in Table IV failure was obtained after 2200 hours at 150 C, while another bearing was still operating after 2800 hours at 125 C. Several high-temperature petroleum greases have been tried in this tester and failure has occurred in the range of 50 to 200 hours.

Table IV

Life Test on DC 44 Grease

Bearing Temperature (C)	(F)	Test Method and Lubrication Life in Hours 1750-rpm Tester	1750-rpm Electric Motor	10,000-rpm Tester
115	239	....	3500	....
125	257	....	2500	2800
150	302	2600	....	2200
200	392	450	....	....

These results indicate that DC 44 has about ten times the life of a good petroleum grease at high temperature and high speed and show that, in bearings with steel retainers, DC 44 has as long a service life as it has in bronze retainers.

Silicone Grease DC 33: Even with silicone fluids as grease vehicles, it has not yet been possible to make a grease that has the extreme high temperature performance of DC 44 and an extremely low freezing point. DC 44 will operate well at a temperature of −40 F, but begins to get stiffer at −50 F. Some applications, mostly military, require performance at even lower temperatures, and this can be obtained with DC 33.

DC 33 is a soap-type grease similar to DC 44, but begins to get stiff at temperatures in the order of −70 to −80 F. Bearing tests have shown that its life at high temperature is about half that of DC 44, so that it is still four to five times as heat-stable as the best organic high-temperature greases.

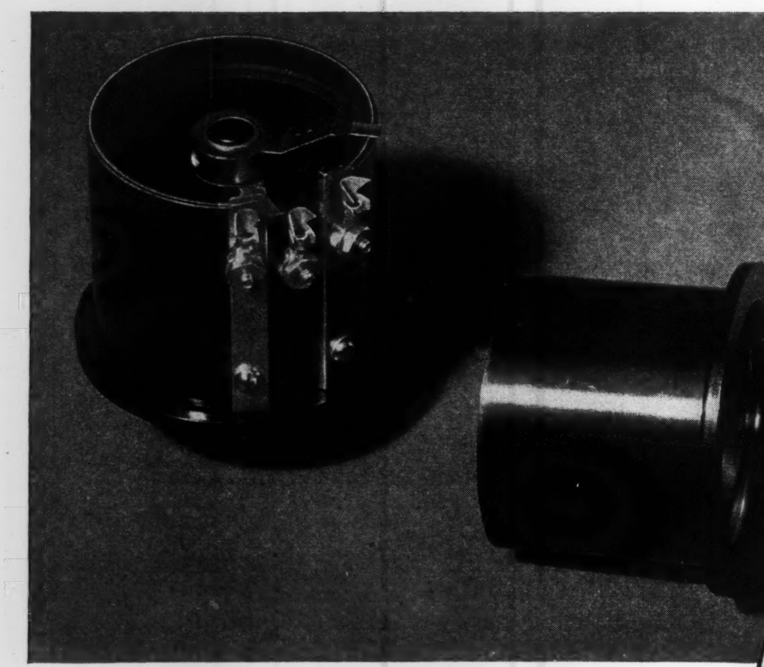
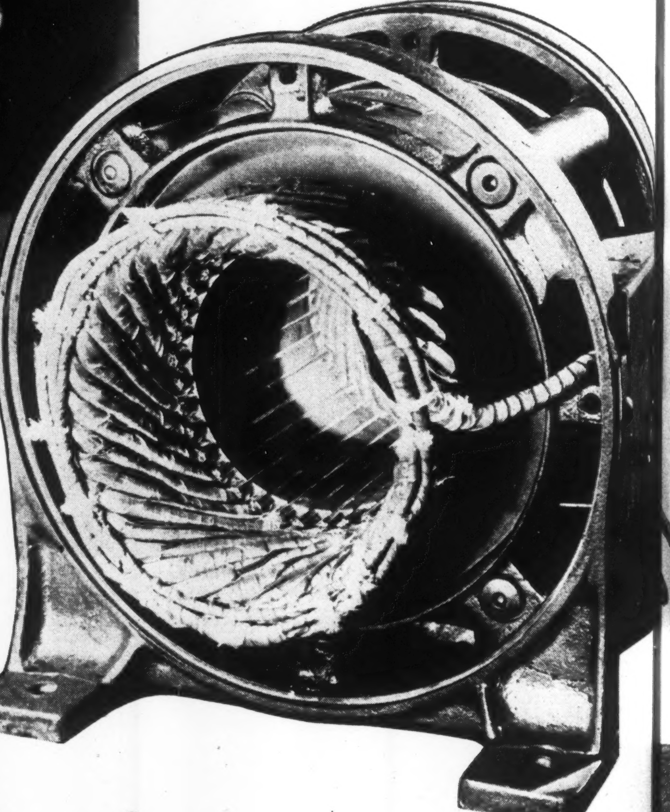


Fig. 1—Left—Molded phenolic potentiometer case supports nichrome resistance wire and serves as mounting for the terminal screws

Fig. 2—Below—Insulating this motor with silicone insulation permitted higher operating temperature, thus greater power output



By John Delmonte
Technical Director
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PLASTICS in Electrical Applications

Factors influencing selection . . . key electrical properties of plastics materials commonly employed . . .

USEFULNESS OF PLASTICS in the electrical field is due to the fact that few materials, if any, are their equal in both mechanical strength and dielectric properties. Then too, while glass, paper, wood, ceramic products, and textiles also have good electrical insulating properties, plastics can be molded or fabricated more readily into intricate shapes.

There is no single plastics material that is pre-eminent in all electrical properties and, in the development of electrical designs embodying plastics, it is necessary to analyze the different properties required and select the right plastics material for the function to be fulfilled. Presumably the designer could consult various tabular data and make a choice of the correct material. However, it usually is desirable to supplement this choice by an examination of the influence of manufacturing procedures and environmental effects upon the properties involved.

A general summary of some of the major electrical applications of organic plastics, together with the forms in which they are used is presented in TABLE I. The numerous significant electrical applications of plastics, cited in the table illustrate the versatility of the materials and also indicate the wide choice of materials available. The list is not complete, although it does record many noteworthy accomplishments. It will be observed that thermosetting molded plastics predominate in many applications. This is to be expected because high-temperature operation accompanies many electrical applications. Illustrative of this point is the manufacture of the molded phenolic potentiometer case, *Fig.* 1. This case serves as a support for a nichrome resistance wire, as well as a mounting for the terminal screws. A brass insert, molded integrally into the base, guides the drive shaft which contacts the tops of the nichrome resistance wire elements. Temperatures cannot go too high however (preferably within 250 F for wood-flour-filled phenolic), although asbestos-base molded phenolics may be operated to 300-350 F. There would be not only mechanical deterioration of the material if the temperature limits were exceeded for too long a period of time, but also a large reduction in insulating strength and a correspondingly higher leakage current and power loss in the material.

Where very high temperature operation is required, the silicone plastics materials are exceptionally well suited. Still too costly for general use, they have, nevertheless, demonstrated operating-temperature limits far in excess of the more conventional insulating materials. In terms of motor performance this has meant a much greater output for a given motor size, because higher operating-temperature limits have been permissible. A silicone-insulated motor stator is shown in *Fig.* 2.

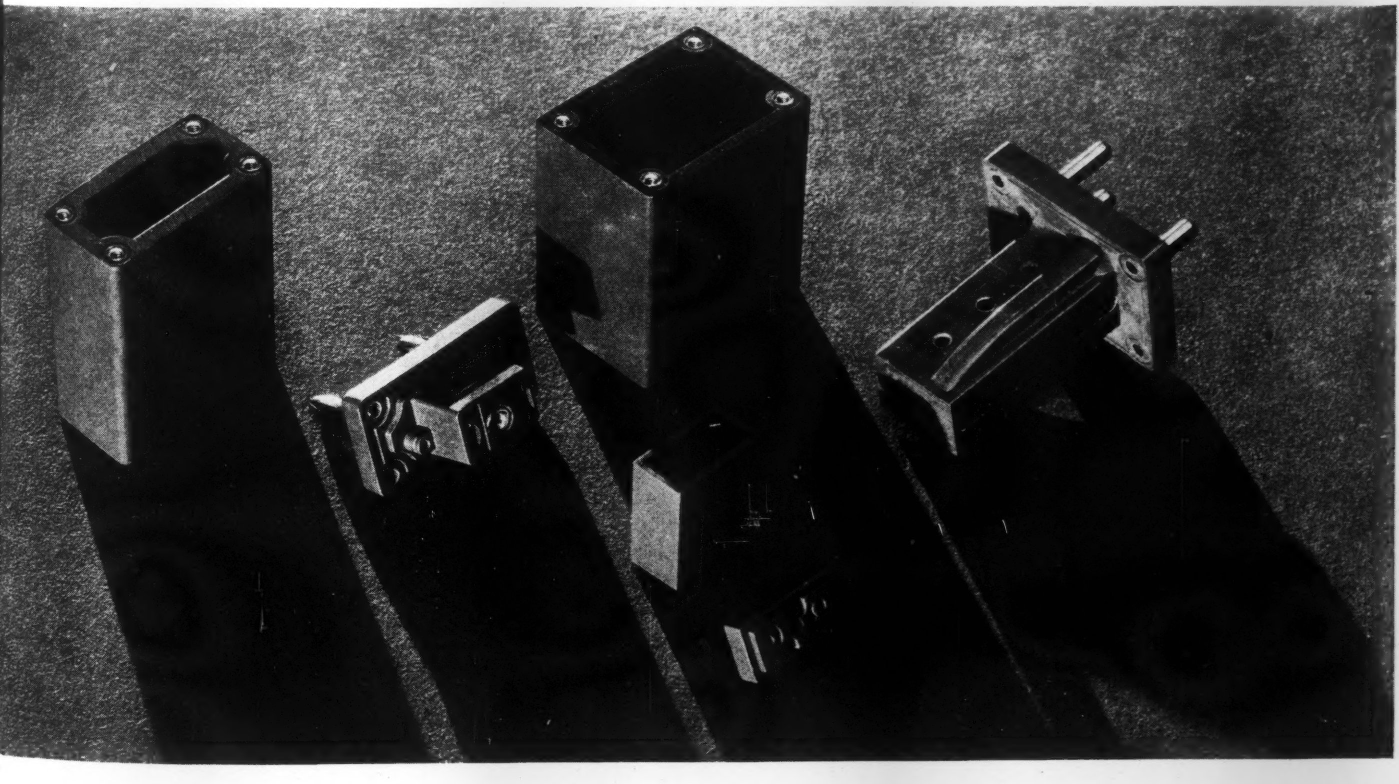
Electrical properties of most direct concern to the designer of machines are: Dielectric strength, dielectric constant, power factor, insulation resistance, and arc resistance.

DIELECTRIC STRENGTH: Ability to withstand high voltage peaks is characteristic of most plastics materials. From 200 volts per mil to 1000 volts per mil, little or no problem is occasioned in the better known materials. While operating voltages usually are safely within the scope of the plastic used, the most severe voltage conditions which may be encountered occur during the proof testing of electrical machines, where agencies such as the Fire Underwriters' Laboratories may apply 1000 plus twice the operating voltage to the apparatus.

Among the phenolic materials, asbestos-base laminates or asbestos-filled phenolics may offer the least dielectric strength; in some instances as low as 50 volts per mil. The general-purpose types are more satisfactory. At the other end of the scale, thin insulating foils of plastics increase in their dielectric strength quite sharply as they become thinner; peak values for most materials occurring in thicknesses between one and two mils. Dielectric strengths of 2000 to 3000 volts per mil are typical values for foils of ethylcellulose, cellulose acetate, polystyrene, and polyethylene.

Higher operating temperatures do not produce a marked fall-off in dielectric strength. There is some decrease, but not enough to cause concern. Other values, such as insulation resistance may change much more sharply. Low dielectric strength may however, be an indication of a poorly molded article. Pronounced weld marks or flow marks in the material may constitute thin air gaps, which have low dielectric strength. The designer should consult with the molder and, if a high voltage stress is present in the final assembly, indicate this feature so that when the mold is designed the possibility of weld marks developing in a region of high voltage stress can be obviated. Weld marks or flow marks and their seriousness are not

Fig. 3—Below—Molded of low-loss phenolic, these crystal holder parts have molded-in metal pins which are sealed air and water-tight at pressure of 30 psi



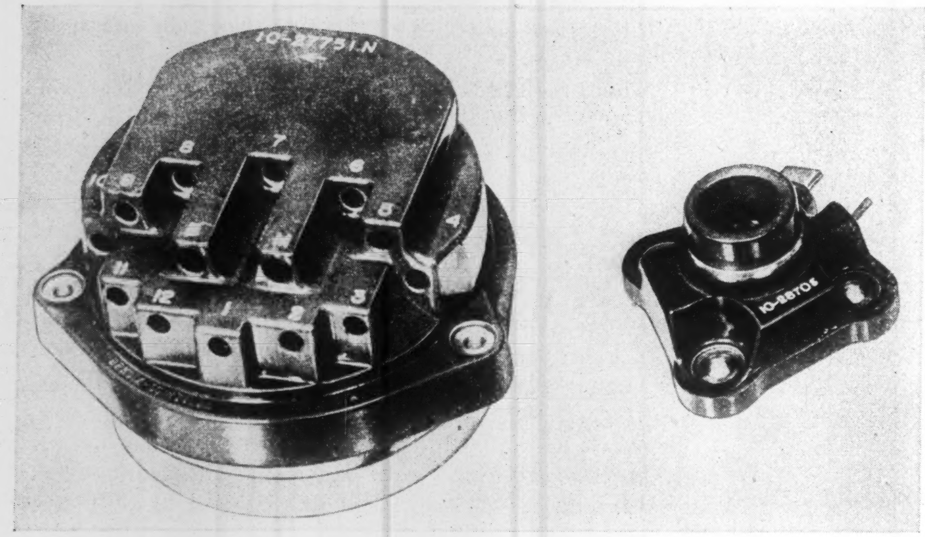
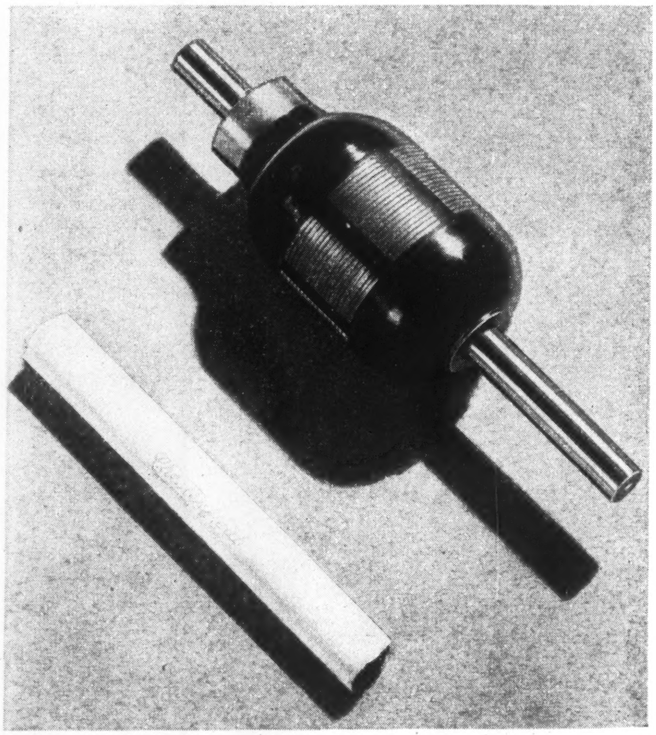


Fig. 4—Left—Molded melamine plastic aircraft distributor head (left) and finger (right) have high arc resistance; permit high service-ceiling operation

Table I		
General Summary—Electrical Applications of Plastics		
Typical Applications	**Materials Used**	**Forms of Material**
Thin foils for insulating purposes, as for motor or relay windings, or in condenser manufacture	Ethylcellulose Cellulose acetate Polyethylene Polystyrene	Foils 1 to 10 mils thick in various widths
Electrical wire insulation	Polyvinyl chloride Polyvinyl formal	Extruded coating or sleeving. Dip coating, thin insulation
Impregnants for motor windings	Phenolic varnishes Silicone resins Asphaltic-base varnish	Liquid impregnants for penetrating windings, baked and dried in situ
Motor housings	Molded phenolic and urea resins	High impact-resistant materials preferred
Radio cabinets	Molded phenolic and urea resins. Cast resins and injection-molded cellulose derivatives	Attractive color assortment available in many standard materials
Insulating bushings and terminal blocks for high-frequency operation	Low-loss phenolics, polystyrene and polyethylene	Molded and sheet forms
Flat insulating washers or terminal strips	Laminated phenolics, sheets and tubes	Grades of material suitable for stamping
Electrical conduit or tubing	Cellulose acetate-butyrate, ethylcellulose	Extruded tubings are held to close dimensions
Ignition distributor housings	Melamine and urea formaldehyde resins	Molded parts, usually with electrical terminals molded into the body
Housings for radar on aircraft	Polyester resins with glass cloth base	Developed into compound curves by low-pressure laminating
Storage battery cases	Hard rubber and asphaltic-base compositions	Thick-wall molded parts designed to withstand battery acids
Connecting plugs and receptacles	Molded phenolics and bitumen-base materials	Bitumen-base parts are for high temperatures
Insulation in vicinity of electric arcs. Switch parts	Shellac, vulcanized fiber, porcelain	Molded or built-up strips
Control knobs for miscellaneous apparatus	Phenolic, urea, or thermoplastics	Colors preferred for identification purposes
Basing cements for lamps	Phenolics, ceramics	Phenolic resin adhesives
Commutator devices	Arc-resistant ureas, melamines, vulcanized fiber	Molded parts with metal strips as contact elements
Welding electrode supports	Heat-resistant and shock-resistant phenolics	Molded with suitable hand grips

Fig. 5—Below—Windings of this small armature are encased in transfer-molded phenolic plastics, thus eliminating expensive dipping and baking of an impregnating coating



always apparent to the naked eye. Their presence can be detected when the molded part is subject to a proof-test voltage. By control of gating or the inclusion of reservoirs to catch the cold flow of material in a mold, the molder can minimize this condition if given complete information by the designer.

Dielectric Constant: Of course the machine designer refers to the dielectric constant of the material only when the material is to be used in electrical applications. From low values of dielectric constant, such as 2.0 for nonpolar materials such as polystyrene, to higher values of dielectric constant, such as 8.0 for the mineral-filled phenolics, the properties of plastics vary.

Power Factor: One of the most important qualities of organic plastics materials is power factor, which is equal to a ratio of the power loss in the material to the product of the voltage times the current through a capacitor when the material is the dielectric. TABLE II lists the power factors of various plastics commonly employed.

With an increasing number of machines being developed for high-frequency operation it is imperative that closer study be given to power factor ratings of the various plastics not only at room temperature, but also at moisture and temperature extremes. One interesting application of low-loss phenolic plastics is the crystal holder shown in *Fig.* 3. In designing this unit, it was necessary to seal the metal contacting pins in such a manner that air and water-tight seals were obtained under 30 psi. The piezoelectric crystals encased in this molded plastics assembly gave outstanding performance.

TABLE II

Power Factors of Representative Plastics

Plastics Material	——Power Factor—— (per cent) 60 cycles	1000 cycles	1000 kc	Dielectric Strength, Short-time (volts/mil)
Phenol-formaldehyde, cast	10-15	1-5	2-5	75-250
Phenolic, molded, wood-flour filler	3-12	3-6	3-5	300-375
Phenolic, molded, fabric filler	10-25	4-10	4-6	200-325
Phenolic, molded, mica filler	1-2.5	.7-1.5	.5-0.9	350-500
Phenolic laminate, Grade X	...	...	2-8	400-1000
Phenolic laminate, Grade C	...	...	2-8	150-600
Urea-formaldehyde, molded, cellulose filler	3.5-10	3.5-5.5	2.7-4.0	300-400
Melamine, molded, mineral filler	7-17	...	4.1	390
Polymethyl methacrylate, sheet	5-6	5-6	2-3	500
Polystyrene, molded	.006-.050	.005-.050	.01-.050	500-700
Cellulose acetate, molded, flow: 286 F	1-6	1-6	1-10	250-365
Cellulose-acetate-butyrate, flow: 286 F	1-4	1-4	1-4	250-400
Polyethylene, sheet	.03-.05	.03-.05	.03-.05	400-475
Polyamide, molded	1.4-5.0	2-5	4-7	350-400

While low power factors are desirable for high-frequency applications, because they mean low temperature rise in the material and greatest stability of circuit parameters, the converse is true when general-purpose materials are to be heated by high-frequency means for molding or gluing purposes. In these instances, of course, high-frequency applications are not anticipated, and the molder or fabricator of plastics utilizes that quality of "high-loss" plastics which permit the molding compositions to be heated rapidly in the presence of a high-frequency field. In fact, high-frequency preheating of molding compounds has become standard practice with some molders.

For best performance at high frequencies, materials such as polystyrene, polyvinyl carbozole, polydichlorostyrene, and polyethylene are outstanding. The designer specifies these materials for lead-in bushings for high-frequency transmission lines or antennas, brackets and supports for circuit elements, insulating washers or spacers (as in co-axial transmission cables), housings for precision electrical apparatus, and many accessories used in conjunction with radar equipment.

From a theoretical standpoint, constancy of power factor over a wide range of frequencies may be expected of nonpolar materials such as polystyrene and polyethylene. Others, which are polar, may be expected to change or vary abruptly. In all instances however, a rise in temperature is marked by an increase in power factor. This is particularly observable in heating plastics at extremely high frequencies, say on the order of 30 megacycles per second. Instead of a retardation of heating rate as temperature rises, the rate may accelerate somewhat due to faster heat generation occasioned by greater power loss as the temperature rises. Data on these changes are still meager, as reliance has been placed largely upon empirical observations. It is hoped that those properly equipped for conducting such fundamental measurements will give more data on changes in power factor with temperature, and on the influences of numerous variables of compositions.

INSULATION RESISTANCE: The resistance a plastic material offers to the passage of electrical currents is termed insulation resistance. It generally is comprised of two parts: (1) Volumetric resistance, which is concerned with the passage of electrical currents through the material and which is controlled by the chemical nature of the material, and (2) surface insulation resistance, which bears a close relationship to surface phenomena. The latter characteristic is governed to a large extent by the designer

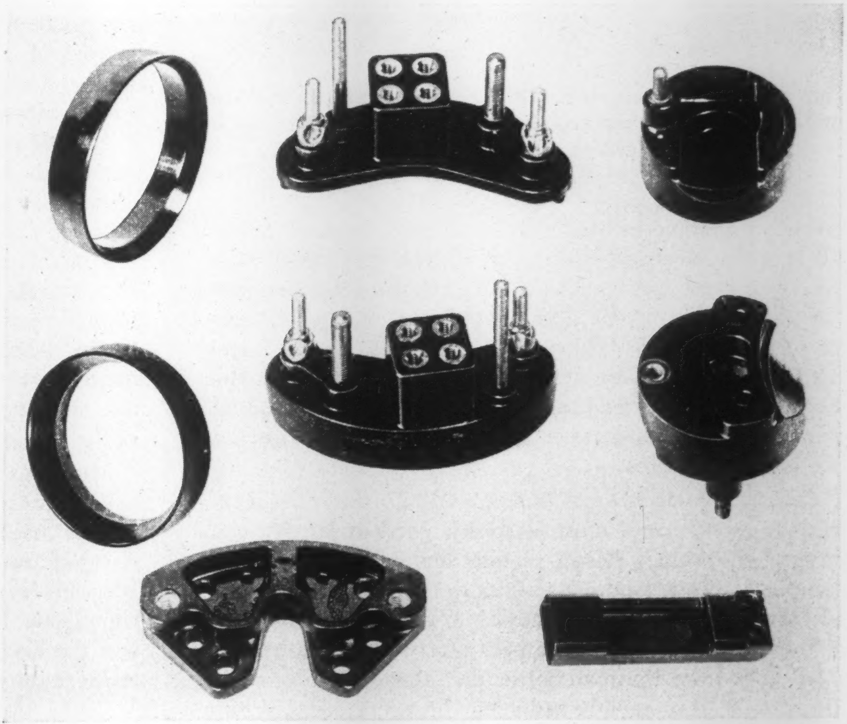


Fig. 6—Right—Interesting electrical machine parts are these molded plastics insulating sleeves, brush supports, wedge rings and terminal blocks which are used in aircraft starter motors

of the electrical machine in which the insulation resistance is significant. Probably no one other single electrical property of plastics materials is subject to as many variables as the surface insulation resistance. The following sums up the effects of these changes upon the performance of a machine.

High Insulation Resistance Means:

1. Very low leakage currents from "live" parts of electrical circuits, and negligible electrical shock hazards
2. Tendency to accumulate static charges and, consequently, dust particles
3. Stability of performance as assured by the constancy of circuit parameters.

Low Surface-Insulation Resistance Means:

1. Electrical shock hazard is present if the body is to touch the machine. Also danger of short-circuiting occurring from a terminal of one polarity to another
2. If the low resistance is a permanent change it may indicate deterioration of the insulation
3. Flash-over tendencies are increased.

Low surface-insulation resistance may be brought on by numerous conditions such as: Changes in operating temperature, changes in moisture content or poor design.

Changes in operating temperature: As the electrical machine temperature rises during continued operation, overall insulation resistance falls off to a marked extent. All tests for electrical leakage should be performed when the equipment has reached full operating temperature. For example, paper-base laminated phenolic which may have a volume resistivity of 10^8 megohm-cm at 20 C, may fall to 10^5 megohm-cm at 100 C. Plastics are not unique in this effect, inasmuch as inorganic insulating materials behave in a similar manner, with decreasing resistance at higher temperatures.

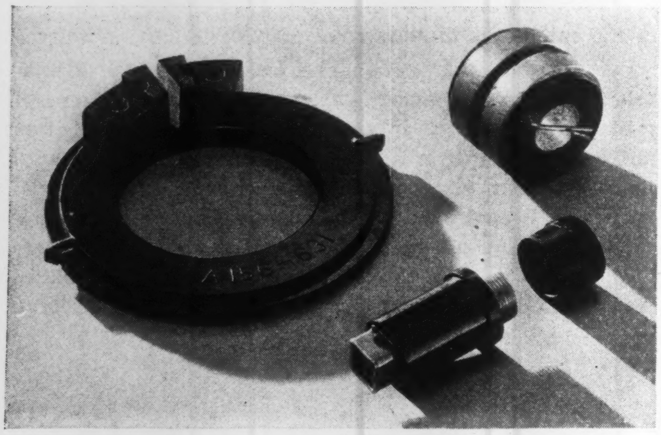


Fig. 7—Brush rocker ring, slip ring and brush holder are typical motor parts molded of plastics. Note the molded-in metal insert in the brush holder

Changes in Moisture Content: While the surface insulation resistance of most plastics is good under dry conditions, the values become quite low when exposed to damp air. Test conditions of 97 or 100 per cent relative humidity generally are employed to explore the behavior of the insulators. The absorbed moisture may either dissolve salts from the material or dirt already deposited on the surface, thus greatly reducing the resistance. Within a few hours the insulation resistance of phenolics, for example, will fall from say 10^7 megohm-cm to 10^3 megohm-cm, and even as low as 10 after several days at the high humidity. Under the right temperature and humidity conditions fungi growths may take place in the films formed on the surface. Materials such as fused quartz, which may have high resistivity when dry, have low resistivity when wetted, due to the fact that moisture forms a continuous film on the surface. On ceramics, however, the presence of a thin film of silicone-resin oil will do much to break up the continuity of the moisture film. Some materials such as cellulose-acetate-butyrate have been reported as being surprisingly good at high humidities, largely because the moisture breaks up into small separate globules, with the result that its surface insulation resistance is not greatly reduced. Polystyrene, polydichlorostyrene, silicone rubber, and tetrafluoroethylene are reported high up on the list with cellulose-acetate-butyrate, all possessing equilibrium surface resistivities of 10^5 to 10^7 megohm-cm after several days at high humidity.

Design Is Important

Poor Design: An inadequate design of an electrical machine may result in a rapid reduction of surface insulation resistance. This implies that spacing metal inserts or terminals too closely together is unwise, particularly if there is a small pocket in which dirt, grease, or dust can accumulate. One or more slots, grooves, or baffles may provide a practical design advantage, and greatly increase resistance between electrical components in a given machine. The designer would do well to remember the reasons for the "skirts" on high-voltage transmission lines, and how these insulators effectively reduce leakage losses to the transmission tower.

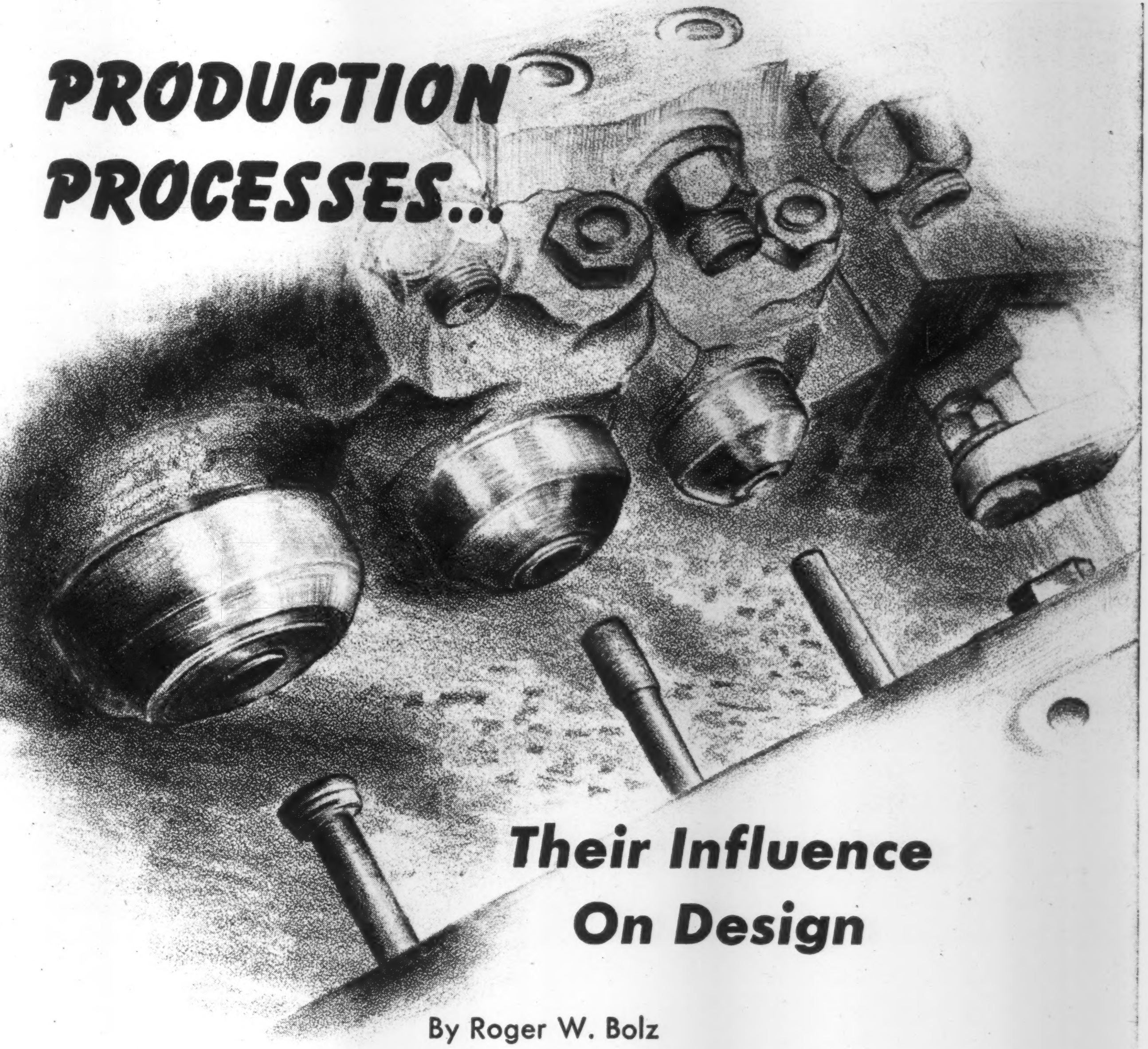
ARC RESISTANCE: Many plastics are unable to withstand arcing without carbonization or scarring across the surface. How serious the damage becomes depends upon the intensity of the arc, and the current going through. The tendency to track across the surface is greatly increased at reduced pressure. In fact, only the molded urea or melamine plastics have been acceptable as ignition distributor housings at high altitudes in the operation of aircraft engines. Most phenolics have failed at the extremely high service ceilings expected of military airplanes. A typical melamine distributor housing is illustrated in *Fig.* 4.

From a design standpoint the inclusion of vulcanized fiber inserts will do much to eliminate carbonization of phenolic parts which do not possess good arc resistance. In the design of electrical switches or circuit breakers, it is important that every care be taken to eliminate materials with poor arc resistance from the immediate vicinity of the arc formed when the circuit is opened.

MISCELLANEOUS ELECTRICAL APPLICATIONS: There have been recent developments in plastics which will influence machine designers in their applications of plastics. A significant one, involving a transfer molding operation, appears in *Fig.* 5. The armature windings have been completely encased in transfer-molded phenolic plastics, and the expensive operation of dipping and baking-on an impregnating coating is eliminated. A procedure such as

(*Concluded on Page* 172)

PRODUCTION PROCESSES...



Their Influence On Design

By Roger W. Bolz

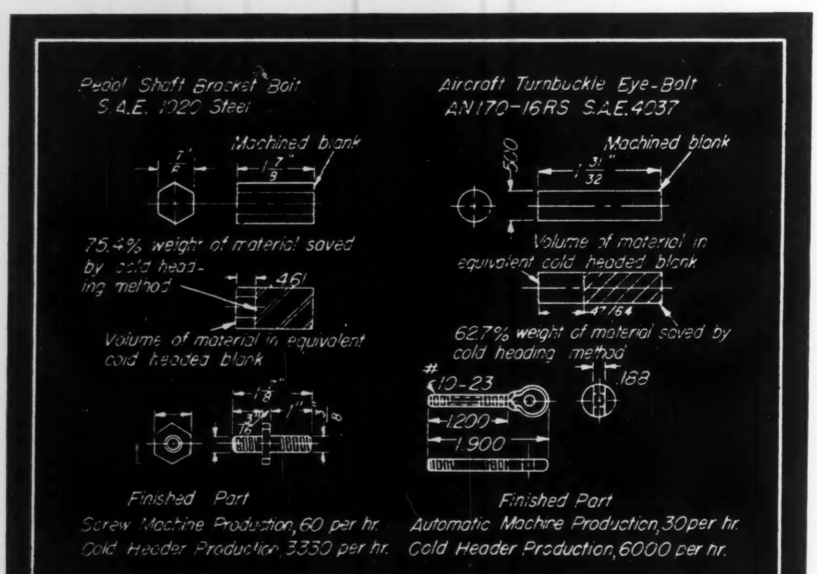
Associate Editor, Machine Design

Part XIII—Cold Heading

AUGMENTING the other high-production processes such as forging, die casting, stamping, etc., available for the mass production of machine parts is cold heading. The many economies and advantages possible by this method of fabrication make consideration imperative for machine parts which fall within its scope.

Cold heading is an old but in many respects a somewhat obscure process. The first automatic cold header was invented, designed and built by W. E. Ward in 1847. However, for many years these machines were relegated solely to the production of common bolts and screws and, as a consequence, were and often still are called bolt headers. Today, though, many types of parts produced on such machines have little relation to bolts or screws, *Fig.* 1. In fact, many parts now cold headed can be produced economically in no other way. Developments and refinements in the process during recent years have been made primarily through improvements and quality of materials, better machines and ingenious die design.

Differing from ordinary forging in that the metal—actually a wire—is upset by dies moving along its axis much as in hot upsetting rather than normal to it, cold heading is desirable whenever practical since


Pedal Shaft Bracket Bolt
S.A.E. 1020 Steel
Machined blank
7"/8
1 7"/8
75.4% weight of material saved by cold heading method
.461
Volume of material in equivalent cold headed blank
1 7"/8
3"/16
1"
3/8
Finished Part
Screw Machine Production, 60 per hr.
Cold Header Production, 3330 per hr.
Aircraft Turnbuckle Eye-Bolt
AN170-16RS S.A.E.4037
Machined blank
.500
1 31"/32
Volume of material in equivalent cold headed blank
47/64
62.7% weight of material saved by cold heading method
#10-23
.188
1.200
1.900
Finished Part
Automatic Machine Production, 30 per hr.
Cold Header Production, 6000 per hr.

the expense of heating and subsequent scaling is eliminated. In addition, a particular advantage lies in the fact that, owing to the change in grain structure produced by the cold working, parts have a greater factor of strength over those of similar dimensions which have been turned, milled, or otherwise machined. All factors being equal, when headed and machined parts which have the same tensile strength are compared, the cold-worked part is found to possess increased fatigue and shock resistance. With identical materials the cold-headed part has, in addition to increased fatigue and shock resistance, a tensile strength 20 to 25 per cent greater. Materials whose strength is increased greatly by cold working—such as the various nonferrous metals and low-carbon steels—are thus considerably improved without extra operations. The smooth, compressed, toughened surfaces produced by drawing raw metal into wire are retained.

Fig. 1—Left—A few of the great variety of machine parts produced by cold heading

Cold heading lends itself to mass production and because it is a fast operation a great deal of time is saved over that necessary for other methods of production. An equivalent amount of tooling for a headed part, as compared to a similar machined one, would produce a greater number of pieces. Many extra operations are eliminated, and where secondary operations must be performed, parts are easily handled in large quantities by automatic setups.

Other advantages which are characteristic of headed parts are the smooth, rounded corners, built-in fillets, and definite lack of burrs. Coupled with these are often the additional ones of burnished or cold-worked finish, uniform accuracy, economy of production, and versatility of manufacture. Finished portions may be either larger or smaller than the original wire—one portion of the blank wire used may be extruded down as well as another portion headed up.

Design for cold heading can effect great savings in material. A machined part must be made from bar stock, the size of which must equal the largest portion of the blank, whereas a headed part is made from wire which usually equals in diameter that of the shank on the part and is headed up to conform to the largest portion. Consequently, scrap loss in heading may be only one to three per cent while that in machining or

Fig. 2—Left—Comparison of material consumption and production rates

forging may run to 75 per cent, *Fig.* 2.

The scope of the cold heading process ranges far beyond that ordinarily realized. The parts shown in *Fig.* 3 serve to illustrate the diversity in design and indicate the wide range of possibilities. Although special machines which handle wire up to 1½-inch are in use, maximum diameter of wire stock which can be cold headed today in standard machines is one inch. For the great majority of production designs, cold heading is considered up to the ¾-inch wire range; from ⅞-inch on, the trend is toward hot upsetting. The final decision, however, depends upon part design and adaptability. Length of wire handled in automatic machines may run from ¼-inch to around 9 or 10 inches, depending upon diameter. Length of parts produced in what are known as rod headers can be almost any length.

Cold-heading machines are classified as solid-die or open-die headers. In open-die machines, wire is fed through two half dies which, when closed, hold the wire firmly, cut it off to length and then transfer it to heading position where punches in a horizontal ram collect and force the wire to conform to the shape of the die impression. These machines are used for parts with wide limits and those too long for solid dies. Largest open-die machine commercially available today is limited to 1-inch wire.

Solid Dies Most Accurate

A solid-die header has slightly different operational sequence in that wire is fed through a quill and is cut off to length first. This piece is transferred in front of the solid die and the "first blow" punch pushes it into the die until it strikes a knockout pin which acts as a stop. The protruding portion is then upset into the impression in the face of the die by succeeding blows. After the piece is finished by one or more blows the knockout pin ejects it. Such a die, of course, makes possible most exacting tolerances and the extruding operations mentioned previously. Normally, length of shank on parts for solid die production should not exceed 10 times the wire diameter. The largest size solid-die header commercially available today is also limited to a maximum of 1-inch wire.

Generally speaking, cold heading jobs run in excess of 5,000 pieces and most often over 25,000 pieces for economical production. Unless design restricts production strictly to a heading operation as with a pyramid or oddly shaped upsets, *Fig.* 4, an order for less than 1000 pieces would most likely be produced on a hand screw machine or lathe. Between 1000 and 5000 pieces could be handled most easily on a single-spindle Brown and Sharpe without excessive tooling costs. If, however, an order runs over 5000 pieces, say 20,000 or 25,000, then the cost of header dies and punches could economically offset any maintenance cost on screw machine tools—with, of course, due consideration to design and adaptability—and the heading process and tooling would then be useful for up to 40,000 or 50,000 pieces. An economical quantity run for odd-shape machine parts is usually considered as beginning at 50,000 pieces. Common types of headed parts such as bolts and other fasteners usually run upwards to the millions. This is readily conceivable inasmuch as some of the smaller screws can be turned out at speeds as high as 400 per minute. Increase in diameter, complexity, or length to a maximum may slow this output speed to a few pieces per minute.

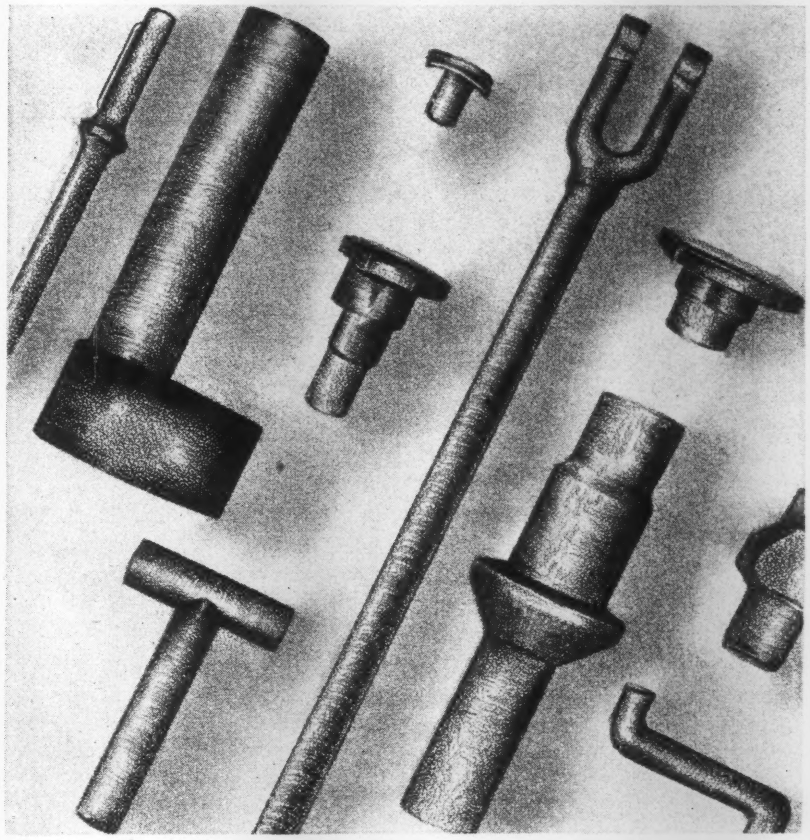
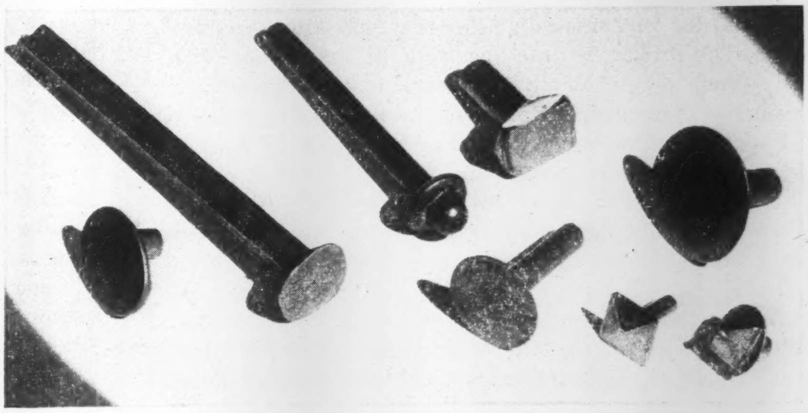


Fig. 3—Above—Sketches of production cold-headed parts which illustrate the possibilities and wide range of design

Fig. 4—Below—For unusual flange or head designs such as these, production by cold heading is usually indicated



Design: A part suitable for heading is usually of simple, and more or less regular contour and, if produced by ma-

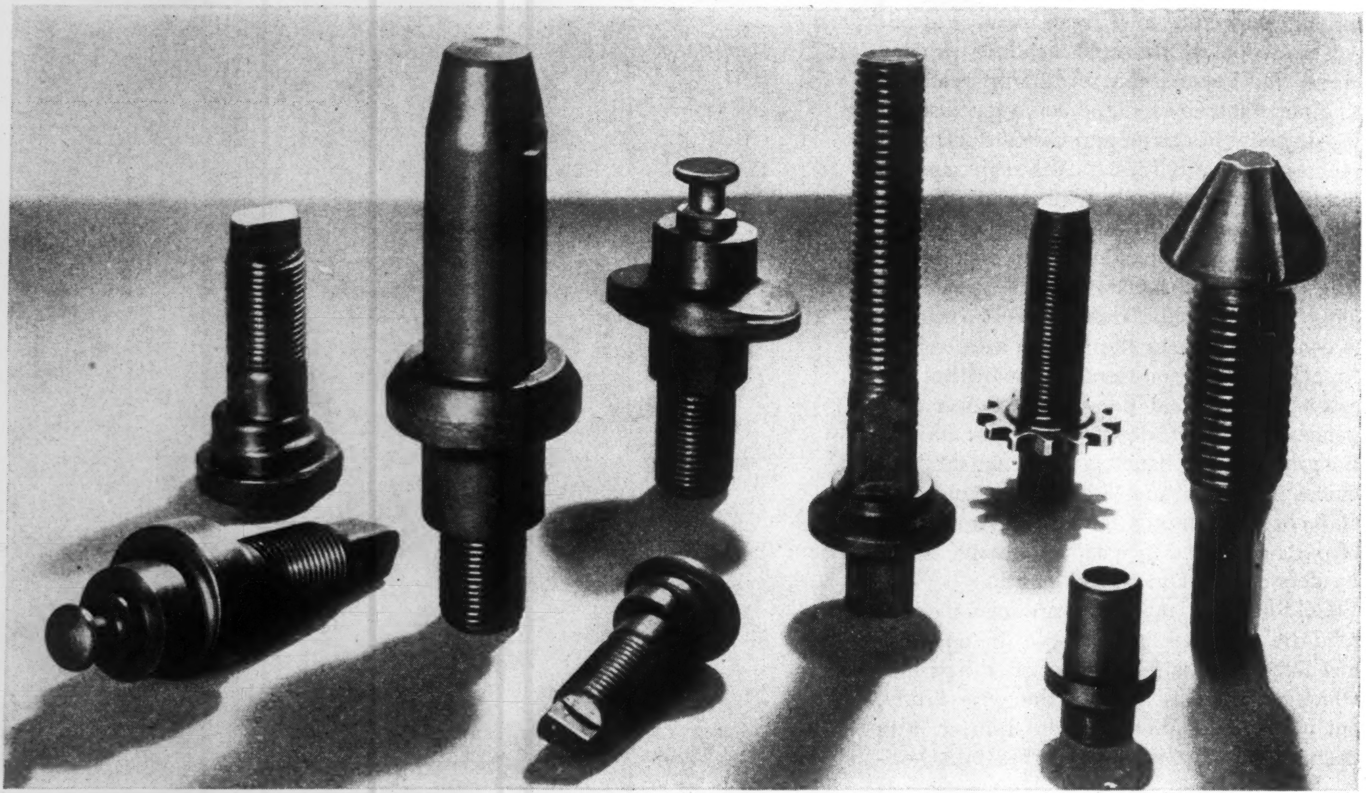


Fig. 5—Ideal parts for cold heading would require an excess of metal removal if produced by machining

chining would require an excess of material removal, *Fig.* 5. Undercuts are seldom present on headed designs unless produced by a secondary operation, *Fig.* 6.

Factors which determine the method of heading are: (1) Relation of the amount of stock required for forming a head to the wire size in number of diameters, to find the number of strokes or upsetting steps necessary, and (2) the relation of the length of a blank to the wire size in number of diameters, to determine the type of die.

Experience has shown that when the length of wire to be upset into a head is less than twice the diameter of the blank, a single-stroke machine is most generally used. If, however, the length of wire to be upset is over twice but less than four times the diameter for the resulting blank, a double-stroke header is used. Where the length of wire to be upset is over four times the blank or wire diameter, either a three-stroke header or a larger capacity double-stroke machine or reheader is used. In each instance, though, these figures may be exceeded if stock is well annealed and of high headability with no long, slender portions. By the same token, with hard wire of poor headability, something less than these figures must be used. Generally, the practical limit for the three-blow header is six times the wire diameter. For some regularly contoured heads which are rounded and not too thin, this figure can be upped to about 10 to 12, though only with material of good headability.

The length of a part determines the type of die. With part lengths of ten wire diameters or less and relatively small size wire, a solid-die machine can be used. However, with blanks over this length ratio, an open-die header is necessary and for the majority of such cases a reputable producer should be consulted, especially with extremely long parts since these often must be hand-fed.

As a general rule, it is well to stay within the limit of four diameters for maximum volume of upset. This sets the limit for head or flange diameters at about three to four times a shank diameter, *Fig.* 7, unless a reheading operation is included. With relatively soft materials such as brass, it is possible to produce a head or flange the final diameter of which is seven or eight times wire diameter, *Fig.* 4. In a few instances, heads as much as 14 times the wire diameter have been produced. This, however, is an exception to the general rule as is the fact that in certain cases as much as 26 diameters in volume have been controlled and successfully upset.

Unless design definitely requires it otherwise, upset portions should be specified as circular for maximum economy. These should be within the foregoing volume limitations with head thickness ranging from around ⅜ minimum to ¾ maximum of the shank or wire diameter.

Radii Should be Generous

Wherever a taper is necessary its small diameter should be outward, *Figs.* 1 and 5, unless special machining is justified. Sharp corners should never be specified—as large a radius as possible will extend die life immeasurably. Generous radii or tapers should be specified for stepped diameter junctions.

Average extrusion work with small reductions can be had with long die life. In fact, some designs in small sizes have been extruded in production with a 50 to 60 per cent reduction. This drops to around 12 to 15 per cent for ½-inch wire. On heavy extrusions, however, drastically lowered die life must often be accepted.

Parts having heads or portions of spherical shape can be produced easily by cold heading and subsequent shaving to size. Necked pieces, likewise, can often be produced by a double operation. The part shown in *Fig.* 8 was upset to form the hollow head leaving a plain shank. In a second semiautomatic open-die header, the 21/32-inch diameter flange was formed while extruding the thread section for rolling. As can be seen from *Fig.* 1, many variations of this method can be utilized in design.

Threads—as can be judged from the foregoing illustrations—should, whenever possible, be specified as rolled. Thread rolling is a highly economical adjunct to the cold heading process and will effect, if specified, lowest possible cost in the manufacture of threaded parts.

MATERIALS: In considering suitable materials, it is well to remember that those which are ultrasoft, such as pure magnesium, or ultrahard, such as hardened steels, do not make for good headed parts. Perhaps the best headability characteristics are possessed by wire material having medium hardness and excellent cold-working properties. This would imply a relative absence of free-machining elements such as sulphur, lead, phosphorus, silicon, etc., from ferrous materials. Sulphur in amounts up to 0.050 per cent has little effect, but over 0.050 per cent it reduces the ability for cold work. Alloying elements such as manganese, nickel, chromium, molybdenum, and vanadium also reduce the capacity for cold working of ferrous materials.

General overall specifications for ferrous heading wire may be outlined as follows:

1. Elongation 20 to 45 per cent
2. Reduction of area 50 to 70 per cent
3. Tensile strength 70,000 to 90,000 psi
4. Wire free from seams, inclusions, etc.

Although neither the elongation nor the reduction of area alone can be considered an indication of headability, the two together are useful. If the elongation is less than that specified above, the wire is not sufficiently plastic to fill out the die impressions properly. Cracks usually form before complete upset. Similarly, if the reduction of area falls below the specifications, the wire can be headed, but only poorly, and die life decreases greatly.

TABLE I*

Approximate Composition of Materials for Cold Heading

Type of Steel	C	Mn	Si	Cr	V	Mo	Ni
Carbon	0.08	0.20	0.05				
	0.15	0.45	0.05				
	0.18	0.75	0.05				
	0.20	0.60	0.05				
	0.35	0.80	0.15				
	0.40	0.80	0.15				
Manganese	0.35	1.35	0.20				
	0.35	1.65	0.20				
Chromium	0.38	0.60	0.20	0.90			
Chromium-Vanadium	0.38	0.65	0.20	0.90	0.15		
Chromium-Molybdenum	0.38	0.70	0.20	0.90		0.20	
Molybdenum	0.40	0.80	0.20			0.25	
Chromium-Nickel	0.38	0.70	0.20	0.60			1.25
Nickel	0.35	0.70	0.20				3.25
Nickel-Molybdenum	0.40	0.75	0.20			0.15	1.20

*From "Hot and Cold Heading"—Alfred S. Jameson, *Metal Progress*, Nov. 1940.

Tensile strength specification is necessary not only to insure a part which fulfills requirements but also as an indication of resistance to upsetting. When the tensile falls below the minimum, the wire will be so soft that, although it can be headed, sponginess may result with attendant heading difficulties. Where the tensile must necessarily be over the maximum specified, special coatings must be cold-drawn into the wire to assist the upsetting.

Specifications for nonferrous heading wire are similar but the tensile strength need have no particular low limit. However, with extremely low tensile strength wire, hot heading or a preheat operation must be resorted to.

The common and usually most economical heading stock is probably ferrous wire with a carbon content of 0.10 to 0.15 per cent. Most easily worked steel is that with less

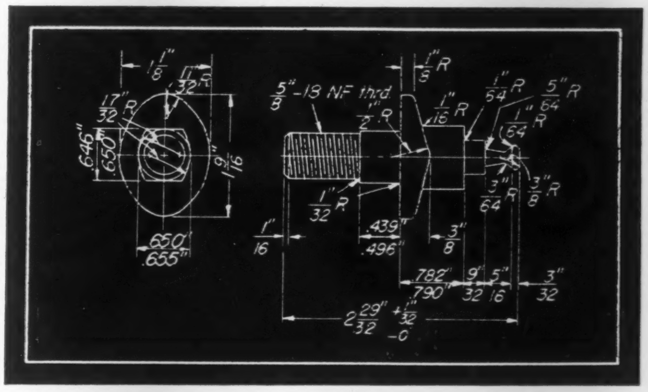
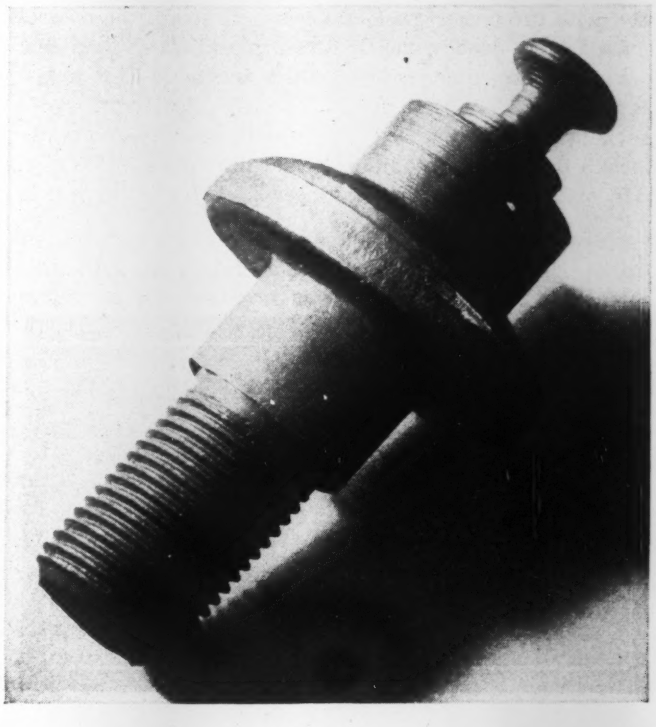


Fig. 6—Elliptical collar, cold headed in two blows from 5/8-inch SAE 1035 steel and sheared to size, would be difficult to produce otherwise. Undercut head portion requires a secondary operation



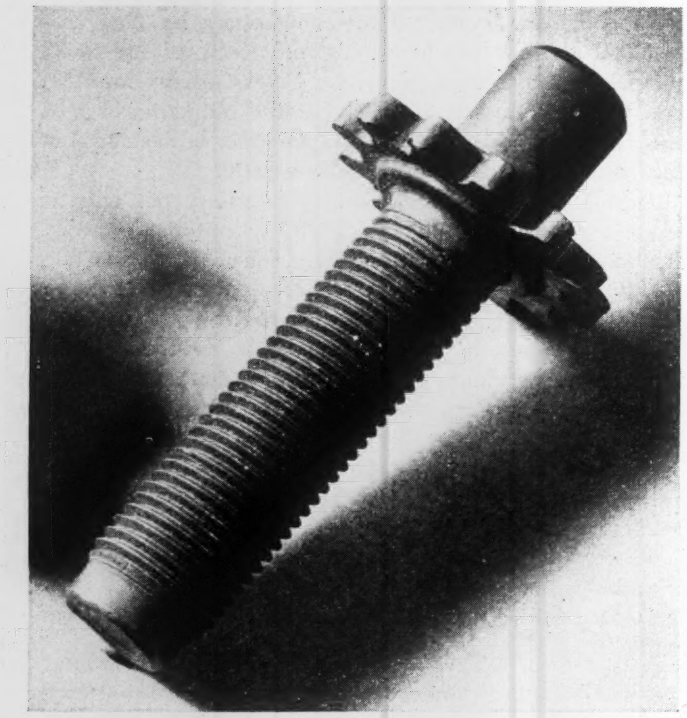


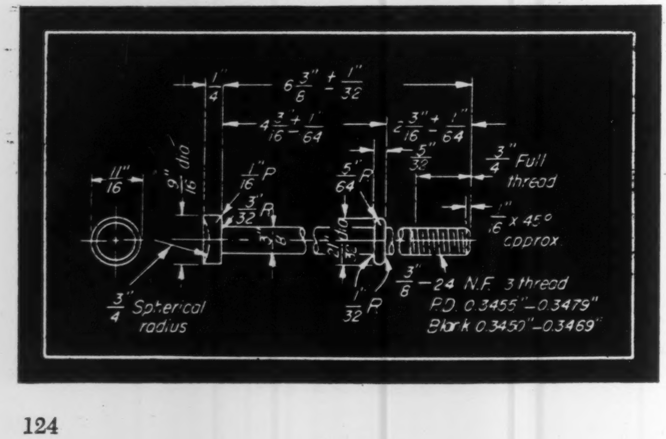
Fig. 7—Brake adjustment unit. Cold headed from ½-inch SAE 1035 steel, a flange 0.130 to 0.150-inch wide is upset to 1⅛-inch diameter. Threaded end is extruded to size for rolling and serrations blanked in secondary operation

than 0.12 per cent carbon, but this material is subject to excessive grain growth. Used widely for fasteners of all types is a 0.15 to 0.25 per cent carbon steel. Carbon steels ranging from 0.25 to 0.45 per cent are specified where maximum physical properties are desired by subsequent heat treatment. Thus, SAE 1035 steel is most commonly employed for such parts.

Coarse-grained steels—1 to 5 McQuaid-Ehn test—in the lower carbon ranges can be used for parts under 7/16-inch diameter which require special physical properties after heat treatment. An oil quench, however, must be used with the coarse-grained steels. Although the coarse-grained wire usually has a better plastic flow than fine grain, the fine-grain wire—5 to 8 McQuaid-Ehn test—must be used on parts over 7/16-inch diameter to avoid cracking after heat treat. The usual practice is to water quench these parts to obtain desired properties.

In almost all cases, if the physical properties of a part are specified, a manufacturer can produce or obtain a wire which will be of satisfactory headability and fulfill the requirements after heat treatment. The approximate compositions of the normal run of materials specifically intended for cold heading are shown in TABLE I.

Fig. 8—Necked aircraft part cold headed from SAE 1020 steel wire and flanged in a second open-die machine



Stainless steels of the 18-8, 25-12 and 12 to 17 per cent straight chromium types are often used. The nonferrous materials include 17S and 24S aluminum alloys and also various brass alloys with 63 to 100 per cent copper. Most nonferrous alloys are used because of special color, appearance, electrical properties, corrosion resistance, or to reduce weight. Costs are usually higher than for steel, however. Magnesium alloys are generally not suitable. Monel can be used and certain special instrument parts have been made from Invar.

TOLERANCES: Generally speaking, die life is indirectly proportional to the tolerance limits held on parts produced. Experience has shown that limits of plus or minus 0.001-inch can be obtained and in some cases are not unreasonable. Most die-life calculations, however, are based on plus or minus 0.002-inch, a tolerance usually maintained on regular headed blanks. Reducing this to plus or minus 0.001-inch would cut average die life by 50 to 95 per cent.

Exacting Tolerances Practical

Experiment has shown that where highly exacting limits are necessary, special honings of the dies, use of tapers, fillets, and special die designs can be resorted to for increasing die life to a practical length.

Based on expected die life with limits of plus or minus 0.002-inch, a minimum of 250,000 pieces can be produced using 0.10 to 0.15 carbon steel, brass, or aluminum wire. This figure drops to around 125,000 pieces for steels such as SAE 4037, or 50,000 to 70,000 pieces for stainless steel or Monel wire, etc. If the tolerance can be increased to plus or minus 0.004-inch, which satisfies most general run designs, the normal 250,000 minimum figure can be increased to well over 300,000.

The foregoing figures are, of necessity, based upon solid-die work. It is usually found that open-die work automatically means increased die life by several times due to use of four impressions in the die as compared to one or two in a solid die and, in addition, freedom from friction upon ejection, etc. The open-die header, however, will not hold the exacting tolerances possible with a solid-die machine and consequently is used primarily for those parts that will require further machining operations. Extrusion work cannot be done on open dies.

Tolerances for the overall length of parts should be as liberal as possible. Close limits can be held on the die upset section but overall length should be at least plus or minus 1/32-inch or more wherever possible. Plus or minus 1/64-inch and in certain cases 0.010-inch can be had but should be considered as less economical in production.

Collaboration of the following organizations in the preparation of this article is acknowledged with much appreciation:
Allied Products Corp. (*Figs.* 5, 6, and 7) .... Detroit
American Brass Co. .... Waterbury, Conn.
American Institute of Bolt, Nut and Rivet Manufacturers
(*Figs.* 1. 2 and 3) .... Cleveland
Bridgeport Brass Co. .... Bridgeport, Conn.
Corbin Screw Division, American Hardware Corp.
(*Fig.* 4) .... New Britain, Conn.
Ford Motor Co. .... Detroit
Lamson & Sessions Co. .... Cleveland
Standard Tube Co. .... Detroit

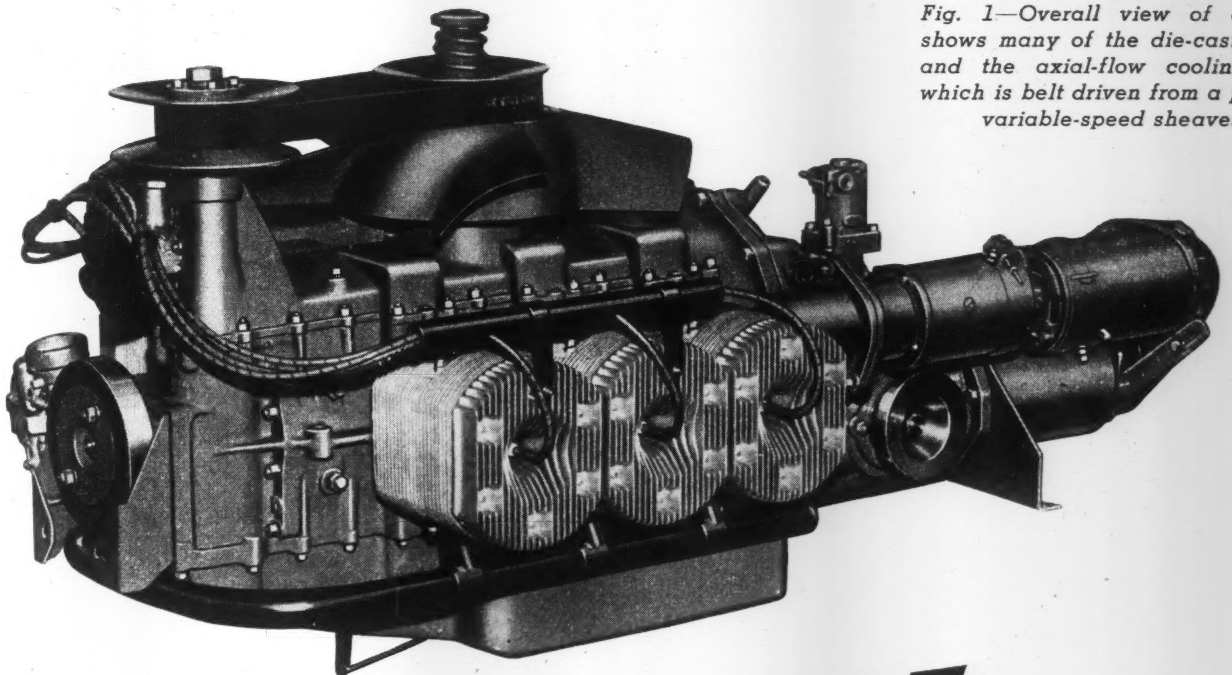


Fig. 1—Overall view of engine shows many of the die-cast parts and the axial-flow cooling fan which is belt driven from a pair of variable-speed sheaves

NEW ENGINE...

incorporates many novel features including slide valves, built-up crankshaft and generous use of die-cast light metals

THE recently developed Jack & Heintz Precision Industries engine, *Fig.* 1, is a six-cylinder, horizontal, opposed air-cooled type which incorporates sliding sleeve valves in contrast to the conventional poppet valves. It consists essentially of several aluminum or magnesium die castings, namely, crankcase, cylinder head, oil pan, accessory cover, and front and rear covers. The two crankcases, *Fig.* 2, are identical, each comprising a bank of cylinders and half a crankcase, and are bolted together on the center line of the crankshaft.

The complete crankshaft assembly, *Fig.* 3, consists of the crankshaft, main bearings, connecting rods, pistons and piston pins, and embodies features not found in conventional designs. For example, the crankshaft is of the built-up type in which the individual parts are positioned by gear-type face clutches and held together by bolts and special nuts, thus eliminating the necessity of using split connecting rods or main bearings.

A newly developed bearing alloy is bonded to the crankpins, main journals and thrust surfaces of the crankshaft. Each connecting rod is an alloy steel forging with a standard bushing pressed into the piston-pin end, the large end being ground and honed to a fine finish. Main bearing blocks are light-alloy die castings into which are fitted nitralloy bushings the inside diameters of which are superfinished.

Pistons are forged of aluminum and piston pins are the free-floating type fabricated from alloy steel tubing, pack hardened, ground, superfinished and fitted with an aluminum button pressed into each end.

The complete valve-shaft assembly, *Fig.* 4, consists of the valve shaft, timing gear and three pairs of supports, valves and eccentrics together with their levers, connecting rods and pins. The valve shaft is driven through a fabric-base laminated plastic gear at one-quarter engine speed by a steel pinion on the crankshaft. Supports are magnesium die castings and provide bearing surfaces for the valve shaft as well as for the pins which locate the fixed ends of the levers. Cast iron eccentrics, integrally counterweighted, are secured to the valve shaft by means of a tapered lock. Their rotary motion is translated into the reciprocating motion of the slide valves by a forged alu-

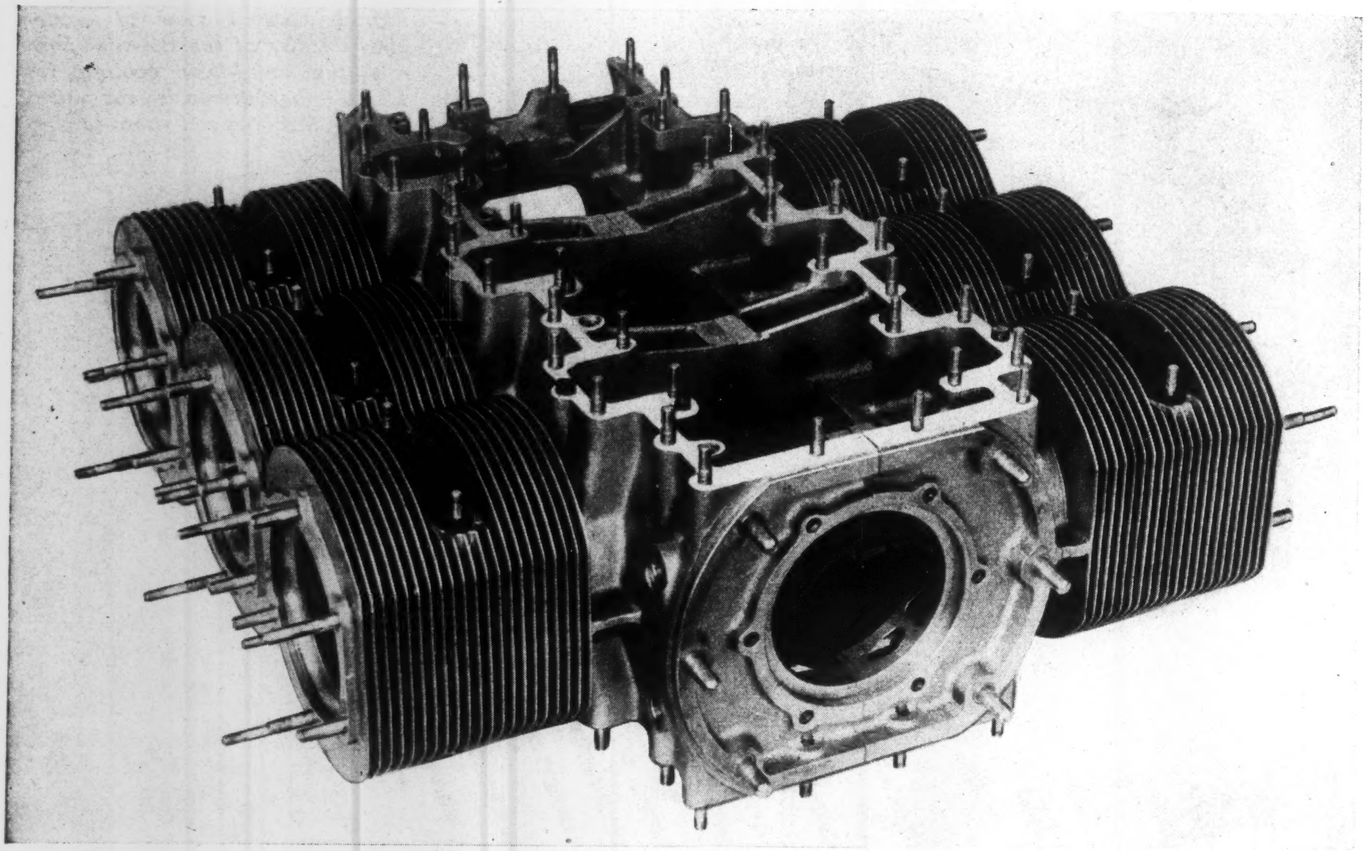
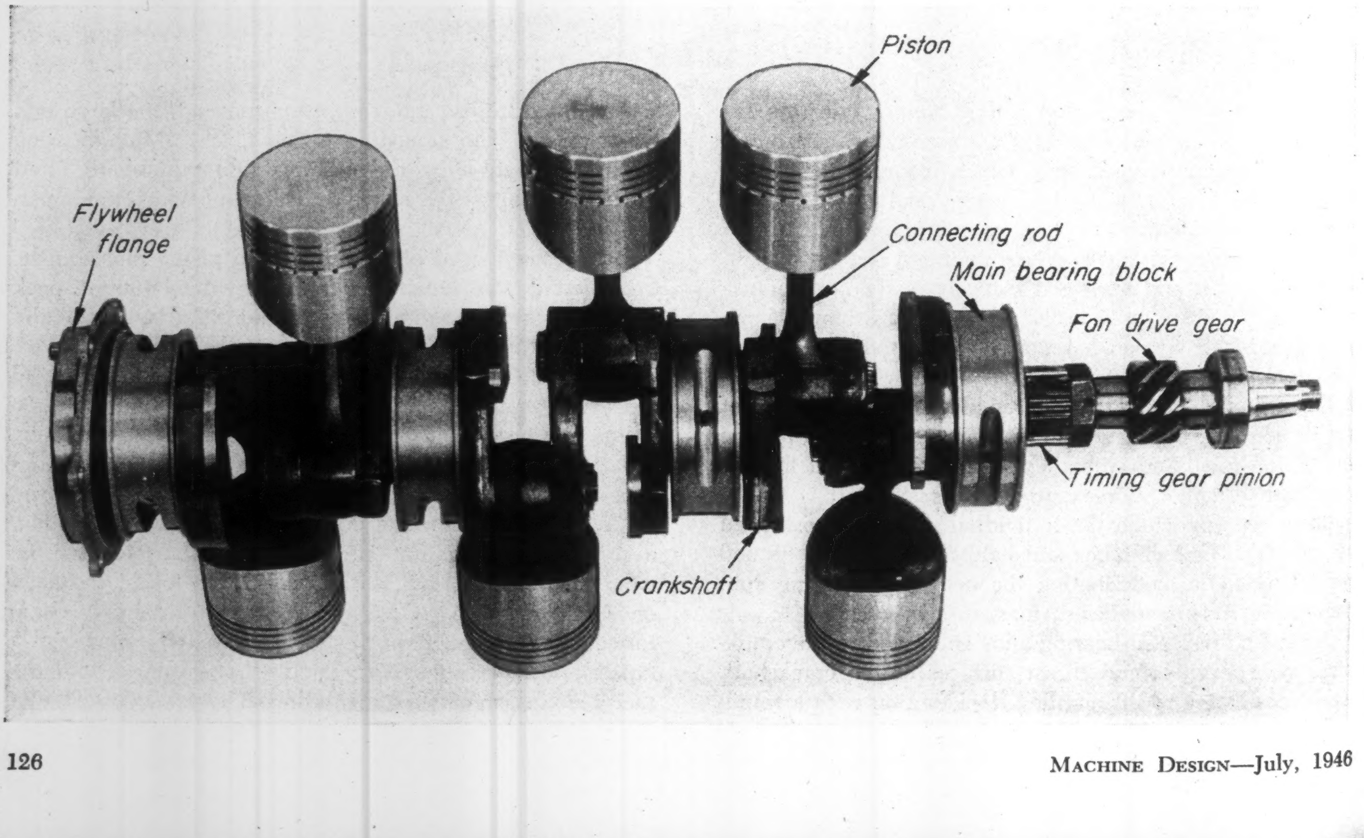


Fig. 2—Above—Two crankcases are assembled by bolting together at the center. Each crankcase is a single die casting and affords a good example of the intricacy of parts that can be die cast when properly designed

minum connecting rod which rides on the eccentric and is attached to the forged steel lever. Levers have close-fitting ball ends which engage the sockets, which are attached to the slide valves by alloy steel rivets. The valves are nitrided alloy steel, formed after heat treatment, then honed and superfinished. The valve opens and closes the port every time it traverses the port in each direction, the

Fig. 3—Below—Crankshaft assembly. Split connecting rods or main bearings are not required because crankshaft is built-up type, its parts being positioned by face clutches and fastened together with special bolts and nuts

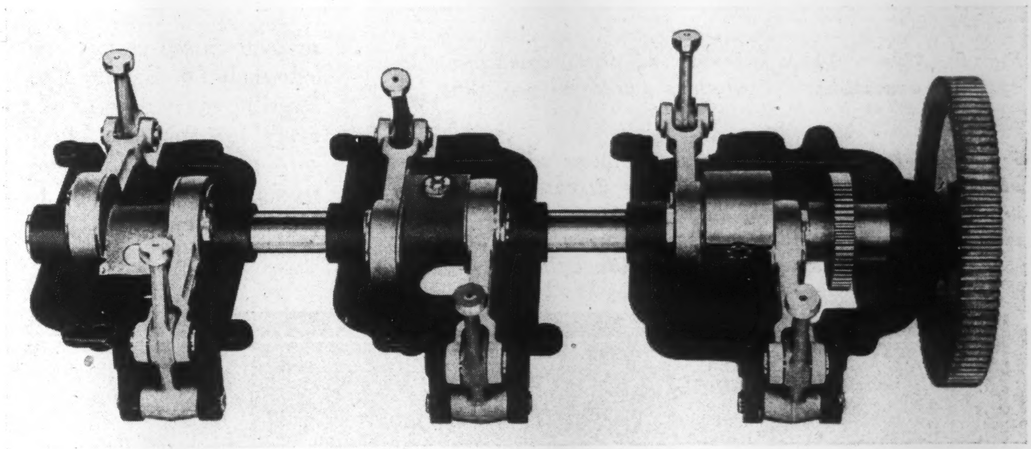

Piston
Flywheel flange
Connecting rod
Main bearing block
Fan drive gear
Timing gear pinion
Crankshaft

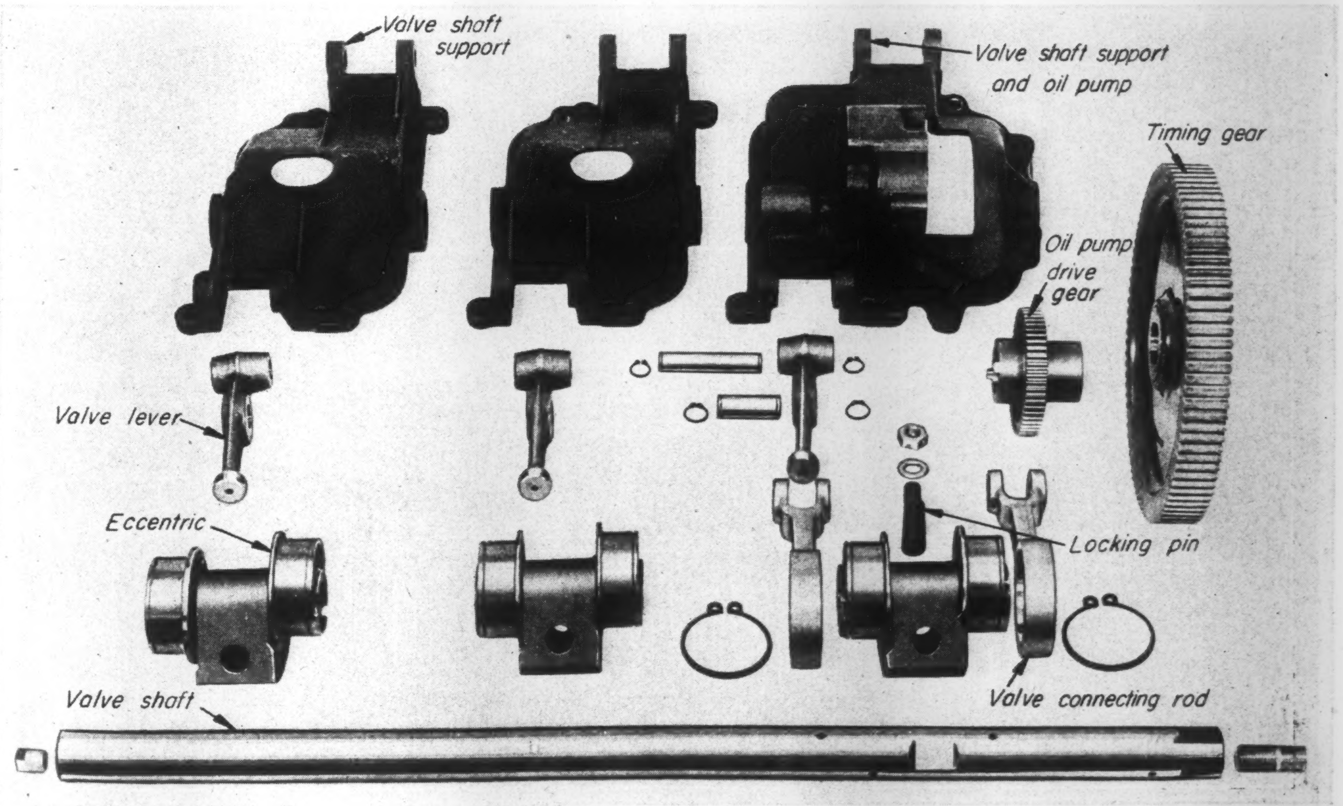
valve port being located so that it travels above and below the combustion-chamber port during each succeeding cycle. Two valve assemblies are employed, one of which drives the intake valves and oil pump, while the other drives the exhaust valves and accessories.

Each cylinder assembly, *Fig.* 5, consists of a pressure spring, reed-type seals, sealing ring, port ring, cylinder, and locating steel segments in addition to the crankcase and cylinder head castings mentioned previously. Each cylinder section of the crankcase has a circumferential groove cut at the lower end and two steel segments are placed in this groove. The alloy iron cylinder has a circumferential shoulder which bears against these segments and the port ring in turn rests on top of the cylinder. The lapped surface of the sealing ring mates with the top surface of the port ring while the other end bears against the pressure spring. When the cylinder hold-down nuts are tightened, the pressure spring is deflected by the cylinder head, imposing a predetermined static load on the complete assembly which actually floats between the two slide valves (see *Fig.* 6).

Three distinct features provide the seal that is required during the explosion cycle in an engine of this design. First, the port ring is split and its original tension permits it to perform like a piston ring for oil control by bearing against both slide valves. During the combustion cycle, the gas pressure forces it against the valves sealing off any leakage that might occur at the ports. Second, the sealing ring is designed to utilize the gas pressures developed during the combustion cycle to increase the force acting on the surfaces of the cylinder, port ring and sealing ring, thus eliminating any leakage at these surfaces. Third, the interference fit between the four thin steel reeds imbedded in the cylinder head and the inside diameter of the

Fig. 4—Right—Valve-shaft assembly. Below—Exploded view of the assembly. Supports are magnesium die castings and eccentrics, of cast iron, are integrally counterweighted and secured to valve shaft by means of tapered lock-pins





sealing ring prevent any gas leakage around the head. These reeds are installed in such a manner that efficiency of the seal increases with increase in gas pressure.

Engine cooling is by axial-flow fan of die-cast aluminum which is belt driven through a pair of variable-speed sheaves. The sheave at the fan is spring-loaded, and changes in the fan-speed ratio are accomplished by a hydraulic servo which is powered by engine oil pressure. Wartime experience on aircraft engines indicated that variations in absolute manifold pressure were practically a straight-line function of the variations in horsepower. This principle was utilized together with an ambient air compensator, to provide the control means for the hydraulic servo. This method of cooling provides relatively constant engine temperatures regardless of climate or power output.

The engine is provided with either a wet-sump or dry-sump system depending on the application. Pressure lubrication is provided for all bearing surfaces and is a definite factor in the long life of all moving parts. In the dry sump system, oil is drawn from the sump by the scavenge pump and is forced through a full-flow filter, then through an oil cooler and into the storage tank. Filtered oil is drawn from the storage tank by the pressure pump and forced through the bearings, the spent oil completing the cycle. A small bleed hole on the discharge side of the pressure pump permits some of the oil to flow through the extremely fine by-pass filter and return to the sump. Theoretically, every drop of oil passes through this by-pass filter every few minutes. In the wet-sump system, the pressure pump draws the oil from the sump and forces it through a full-flow filter to all bearings, the spent oil completing the cycle.

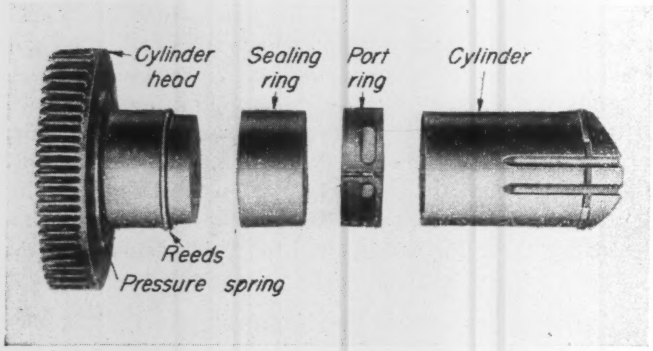
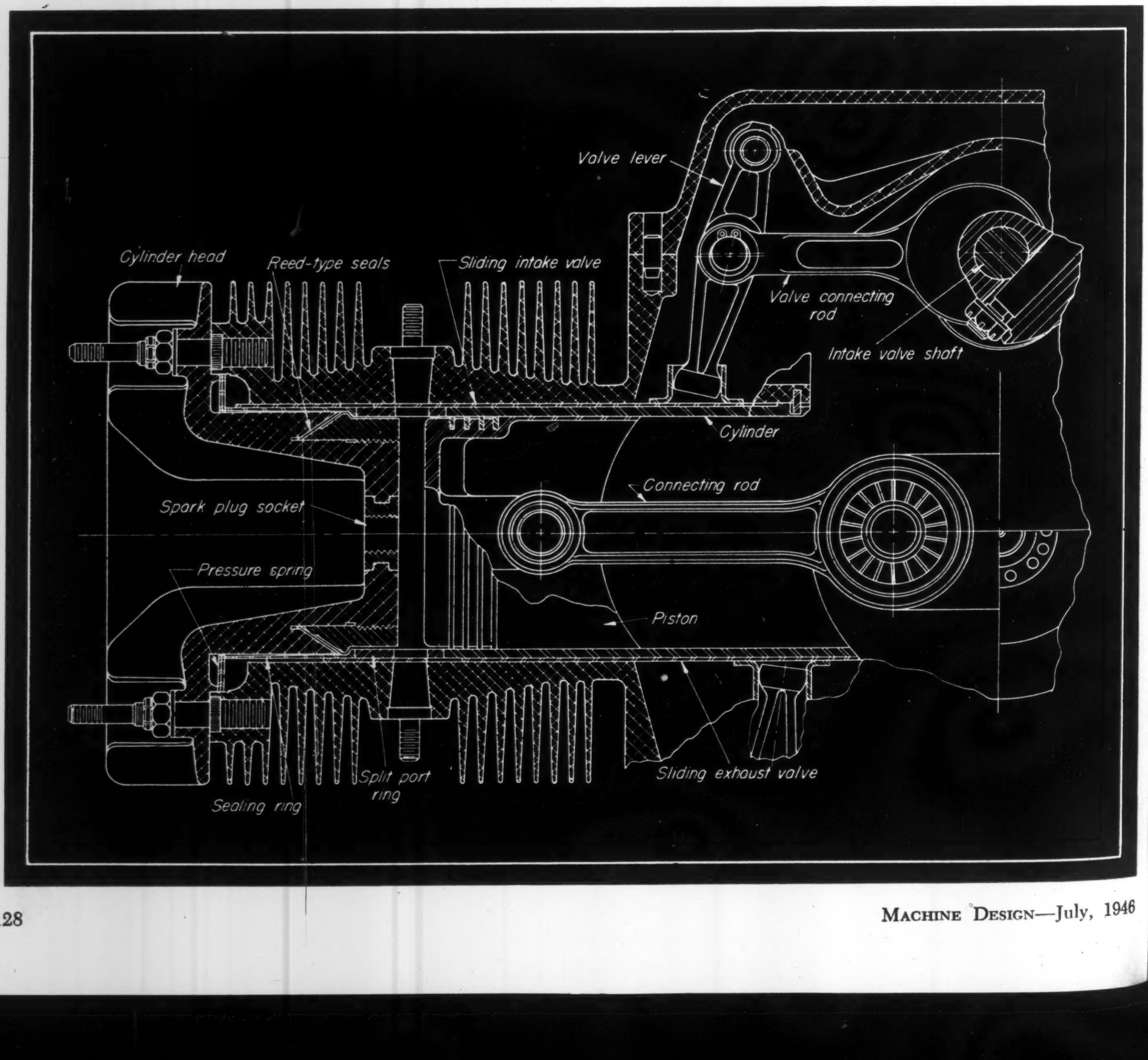

Cylinder head
Sealing ring
Port ring
Cylinder
Reeds
Pressure spring

Fig. 5—Above—Exploded view of parts comprising the cylinder assemblies. Cylinders employed are alloy iron

Fig. 6—Below—Section through cylinder and crankcase shows how sliding sleeve valves are driven off the valve-shaft eccentrics. Also, note arrangement of sealing members such as the pressure spring, sealing ring, reed-type seals, etc., in the cylinder


Valve lever
Cylinder head
Reed-type seals
Sliding intake valve
Valve connecting rod
Intake valve shaft
Cylinder
Connecting rod
Spark plug socket
Pressure spring
Piston
Split port ring
Sealing ring
Sliding exhaust valve

Designing for Precision Casting

By Richard C. Wynne
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Naval Research Laboratory
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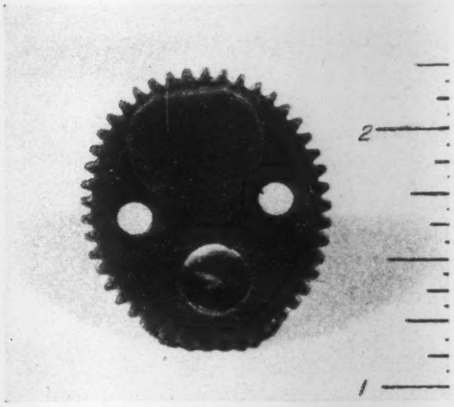
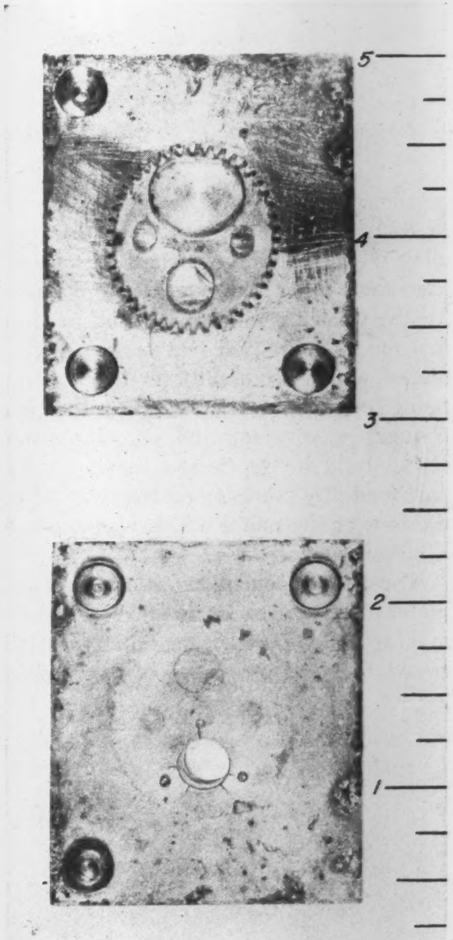


Fig. 1—Above—This elliptical gear cost $100 machined. Duplicate gears were made by precision casting for a fraction of the cost of the original

Fig. 2—Below—Type-metal die for elliptical gear casting shown above



PRECISION casting offers today what is often the least expensive and, in some cases, the only method of making a large range of machine parts. Tooling expense is low and the number of pieces required need not be large to make production by precision casting economical. This last feature is important in determining the potential market for many articles due to the possibility of introducing a limited number before full-scale production is started. Some typical examples of parts made by precision casting techniques are illustrated in this article, and some of the factors which influenced the choice of this process are explained.

Shown in *Fig.* 1 is a small elliptical gear used in a camera sighting mechanism for following the flight of a projectile. The first two gears used in this mechanism were specially cut by an outside contractor and cost the Naval Research Laboratory $100.00 per gear. When several more were desired, the precision casting group turned out 12 beryllium-copper duplicates of the original gears with an expenditure of only 5 man-hours and a material cost of less than $3.00. This included the time and material required for the low melting point metal die shown in *Fig.* 2. The only finishing required was the sprue removal, after which operation the gears were placed in service. They were completely interchangeable with the original machined gears.

STEPS IN PRODUCING precision parts by investment casting were illustrated in our May issue. In the present article several noteworthy examples of precision-cast parts are discussed and a few simple rules given relating to design

The saw bit shown in *Fig.* 3 is the cutting edge of a high-speed mechanical saw that was used in cutting the bullet-infested trees in the South Pacific theatre. Cost of manufacturing the bit by conventional means was 4 cents for carbon-steel bits and 15 cents for high-speed steel bits. These prices could not be beaten by precision casting methods, but forging facilities were so tight that an alternative method of manufacturing had to be found. The parts were produced satisfactorily in carbon steel, but preliminary castings of the high-speed steel bits were unsatisfactory due to the brittle precipitate which formed around the grain boundaries. Later work showed that a different technique was necessary to produce a satisfactory part in high-speed steel. Cost of making the carbon-steel bit by precision casting was 5 cents. Of particular interest in this part were the edges along the back of the bit. These edges were satisfactory in the "as cast" state. The holder for the bits, *Fig.* 4, was also precision cast of low-carbon steel at a cost of approximately 12 cents.

In *Fig.* 5 is shown a small turbine rotor which was used under operating conditions involving high pressure and high temperature. This part was to be made of a heat-resistant alloy, Vitallium, which is virtually impossible to

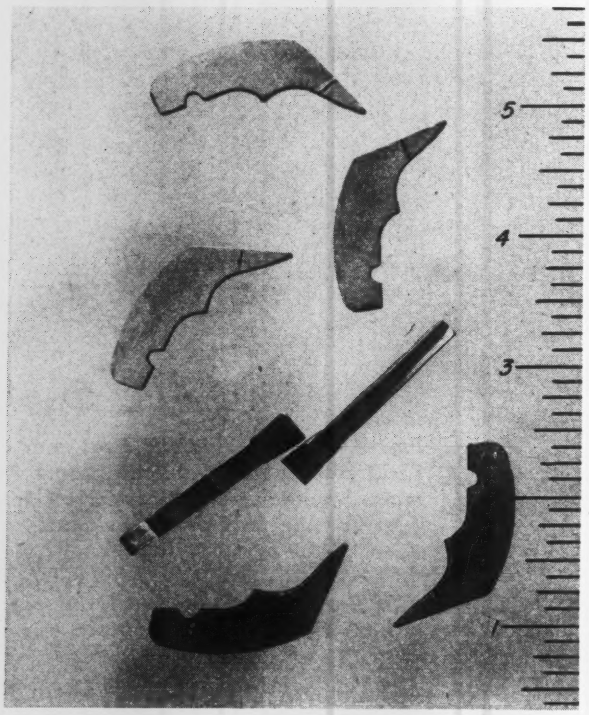
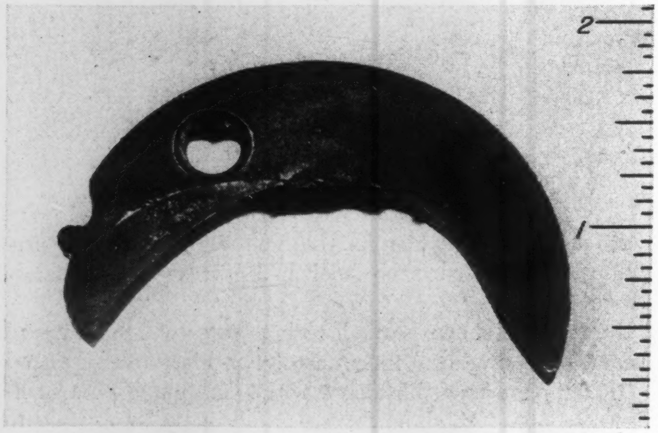


Fig. 3—Carbon-steel bits for high-speed mechanical saw. Precision castings satisfactorily replaced forgings, relieving forging production capacity

machine. An outside contractor submitted a cost estimate of $20.00 per rotor, based on the use of an alternative alloy which could be machined. Even if he could have used some other alloy such a figure was out of the question, for the turbines were to be used in large quantities. The first model submitted had feather edges at point *A* on the blades which were difficult to cast, most of the trouble being encountered in making the wax replica. Some changes were suggested and the rotor was finally cast successfully in the design shown in *Fig.* 5. Cost of the part made by precision casting was 75 cents, not including the cost of the metal which was about $1.50 per rotor for Vitallium alloy.

The Alnico pole piece shown in *Fig.* 6 was for a small motor that was to be made in limited quantities so that making it by powder metallurgy techniques would have raised the cost per piece to a prohibitive figure because of the initial die cost. This part was ideal for precision casting, as the cross section was uniform and there were no extremely fine edges.

Fig. 4—Holder for saw bits shown in Fig. 3 also is precision cast of low-carbon steel. Inch scale shown at right



An experimental microwave antenna is shown in *Fig.* 7. An original model was made by hand and twelve duplicates were desired. A low melting point metal die was made from the original model and twelve parts were cast in a compensated investment which gave castings having the desired dimensions. The castings were sprued as shown and the only finishing required was a turning operation to remove the gates. Total cost of the twelve duplicates was approximately $15.00 including the cost of the die and the aluminum alloy used in making the parts.

Relatively few designers seem to have realized fully the possibilities of precision casting in the light of recent developments in this field. Those who have used precision castings extensively are well aware of their advantages. Others may have a mistaken idea about their limitations. At any rate, many have never stopped to inquire how important such castings might become in their own products if wisely applied.

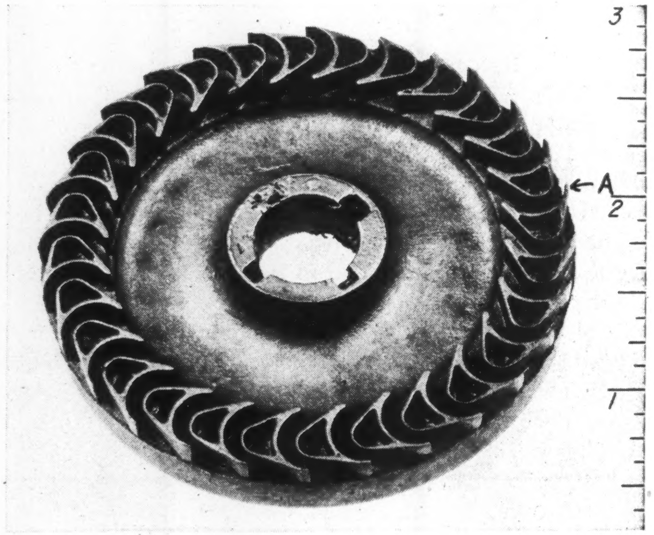


Fig. 5—Redesign of this Vitallium turbine rotor to eliminate feather edges at point A on the blades enabled precision casting method to be successfully used

As in other fields, best results can be obtained by consulting an experienced precision caster before preparing a final design. In lieu of advice from an experienced precision caster, or in developing preliminary designs, the following suggestions will help in designing a part adaptable to precision casting. Actually, there are relatively few unvarying rules of procedure in precision casting.

Wax Pattern Die: Making the die for producing the wax or plastic replicas is one of the most important phases of the process. The dimensions of the replica will usually be the same as the final metal part except where high expansion investments are used, as in the case of the elliptical gear, *Fig.* 1, and radar antenna parts, *Fig.* 7, previously mentioned. In most cases, particularly unsymmetrical parts, the wax will shrink more in some dimensions than in others, and the best practice is to take the original model, make a cast type-metal die and go through the entire process. The resulting casting is measured and then compensation can be made for all dimensional changes occurring in the various steps of the process. A machined die or model can then be made incorporating the necessary changes so that the final casting will have the desired dimensions.

Die Parting: It is desirable for the designer of the casting to foresee where the parting will come and to so design the casting that the parting can be made preferably within a single, rather than several planes. Frequently irregular partings are feasible and may be necessary in many cases, but they are likely to step up both the cost of the replica dies and later cleaning operations. Where possible the parting line should be located so that half of the casting is in each half of the die. Also, the designer should avoid whenever possible a parting that would produce flash at any point where it is difficult to remove. It is good practice to make the parting at a square or other sharp edge of the casting when this can be done readily.

Tolerances: Tolerances should always be specified where they are important, and should never be closer than is essential to meet requirements, because of the extra care involved, and the resulting extra expense. When a critical dimension is across a parting line there must be an added allowance, is most cases 0.001-inch total in addition to the minimum of 0.002-inches per inch that is usually allowed.

Female Threads Difficult

Threads: In most cases either inside or outside threads may readily be precision cast on the piece, but inside threads require that the core forming them in the replica must be carefully unscrewed from the piece. This increases the time required to make the replica, slows production, and increases the cost per piece accordingly. Often it will be found cheaper to cast a hole and tap the metal casting. Regardless of whether or not the threads are cast, blind holes should be avoided whenever possible. Outside threads are usually formed by a split section of the die, usually at the parting line, which results in a slight flash line along the thread of the replica which must be chased before casting, or the thread can be formed by a loose collar having an inside thread not split, which must be unscrewed. This also is a slow operation and, unless there are unusal circumstances such as an unmachinable alloy, it is usually far more satisfactory and economical to machine the threads after casting. Whitworth threads with rounded root and crest and U. S. Standard threads are much easier to cast than square threads because of the tendency of bubbles to collect in the square threads. Approximately 20 threads per inch is about the finest thread that is considered feasible to cast in nonferrous alloys and 14 to 16 threads per inch in ferrous alloys.

Uniform Sections Desirable

Other Factors: Fillets should be employed wherever possible to decrease turbulence as the mold cavity is filled and also to prevent the occurrence of hot tears which may show up around sharp internal corners. Section thickness should also be kept as constant as possible to prevent this same deleterious effect.

Undercuts in cored holes are undesirable not only because of the difficulties involved in the making of the collapsing cores, but also because of the defects that result in the casting. These defects occur in varying degree, depending upon what investing techniques are used, the single investment technique resulting in less breakdown in almost all cases.

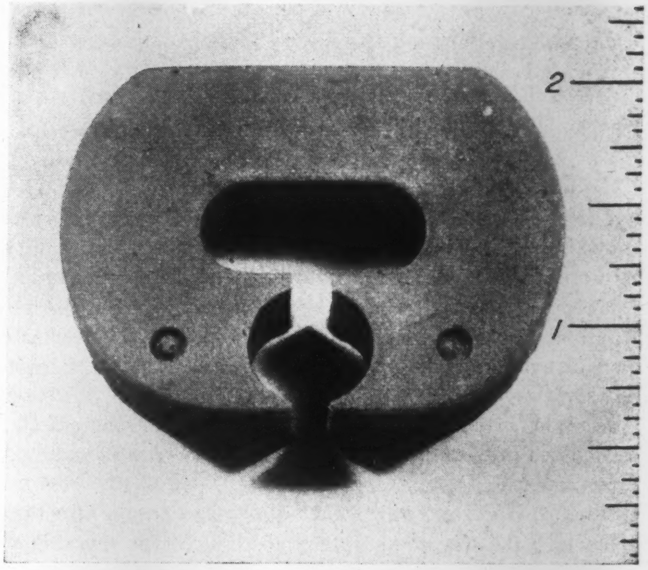


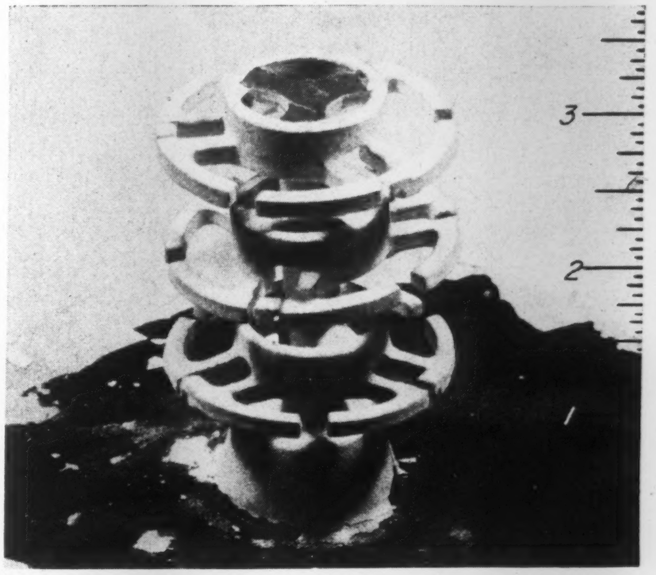
Fig. 6—Precision-cast Alnico hole piece for small motor. Limited quantity production would have meant prohibitive die costs if made of powder metal

Large flat areas should be avoided whenever possible, for the replicas are invariably "dished in" unless a "two-wax" technique is used, which is satisfactory except for the increase in cost which results.

Engraving and lettering of all types can be easily incorporated in any precision-cast part, the only restriction on their use being interference with the extraction of the replica from the die.

In conclusion, it may be said that although precision casting is not a panacea it nevertheless fills an important gap in metal fabrication.

Fig. 7—Experimental microwave antenna, twelve duplicates of which were precision cast in an investment made from an original hand-made model



Stress Gages Si

By Sidney B. Williams*

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Caldwell, N. J.

EXPERIMENTAL methods for verifying stresses and strains under operating conditions are no longer new to most designers, and it is generally realized that loading in a complex structure may be such that a simple strain measurement will not suffice to determine the actual stress. Since structural members are designed on a stress basis it is necessary to have a working knowledge of the stress-strain relation and of the most practical ways of evaluating the state of stress in a test specimen or part such as shown in *Fig.* 1. The method to be discussed is not new in that it concerns the use of a strain rosette, but it does present it in a form that is easier to use and to remember.

When it is desired to measure stress in a single direction at a point, the use of a stress gage reduces the amount of equipment and interpretation required, so that a stress may be measured as directly as a strain. A careful approach, starting from fundamentals, will serve to bring forth the use of the Mohr circles in evaluating biaxial strain data (conditions where knowing a strain does not mean knowing the stress that causes it).

Since Hooke's Law states that strain is proportional to the stress producing it, why not apply the proportionality factor (Young's Modulus, E_y) and compute the stress? The answer is that the strain in any given direction is not all due to the stress in that direction, but may result also (in a lesser proportion) from stresses which are acting perpendicularly, or normal, to the given direction. In a complex structure, strains will generally be due to the combined effects of stresses in both the same (axial) and normal (or lateral) directions. In a simple case such as a tension specimen, the strain in the direction of the applied load is all due to that axial stress, while the *lateral* strains (the "necking down" action) are not due to any stress in the lateral direction (for there is none) but are all due obviously to the applied load and its associated stress, which of course are acting in a direction *normal* to the necking-down action.

Stress, being the average force acting normally to a unit area, cannot act laterally in the simple tension member since the applied load has no component in that direction. Strictly speaking, the word *stress* should be replaced where it occurs in the foregoing by the phrase *normal stress* be-

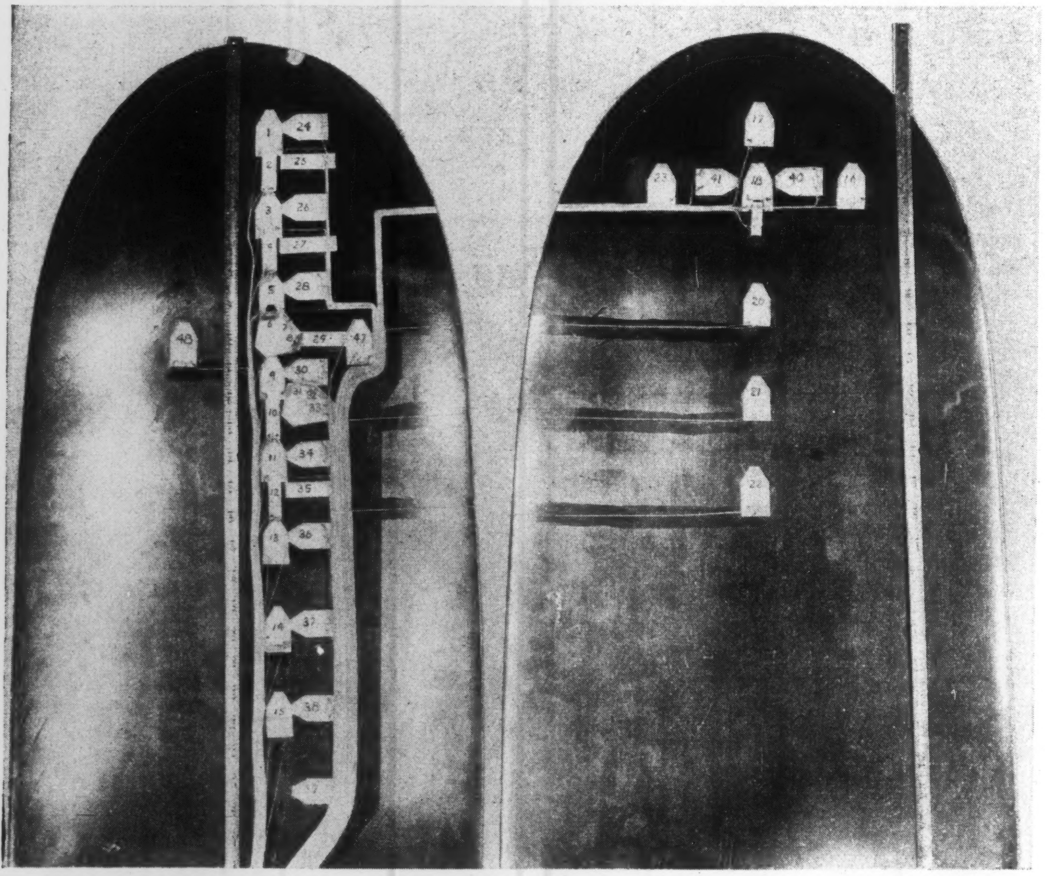

Fig. 1 — Camber and thrust faces of hollow steel propeller, showing stress gages, strain gages, and strain rosettes wired for engine - stand testing. Thrust face carries fewer gages. Stress gages predominate and can be identified by their flatiron-shaped felt coverings

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geSimplify Analysis

BECAUSE strain measurement does not give a direct indication of stress in many cases such as in propeller blades subjected to complex vibratory stresses, the author developed stress gages to aid him in his experimental work. These gages have marked advantages because they aid in solving biaxial stress equations without second measurement or calculation

cause that is correctly applied only to stresses which tend to cause elongation or compression. *Shear stresses* are those which tend to cause angular displacements or rotations. This is complicated by the fact that the one type of stress can hardly exist without the other, and such other peculiar truths that ductile materials loaded in tension fail in shear, but this discussion will be concerned only with the evaluation of stresses at loads well below the ultimate. It will be limited further to the measurement of biaxial stresses at a point—that is, stresses measured on a surface whose curvature is small enough to be neglected, upon which no normal forces are acting, and over whose area stresses and strains vary only with direction. These are assumptions which operate well enough for all ordinary work where gages may be located on a free surface.

Biaxial Stresses and Mohr Circle of Stress

Because there are normal and shearing stresses existing in various directions, measurement is not as simple as measuring a strain and multiplying by the modulus. There are, however, certain features that simplify the situation. These are:

1. Six measurements are required to determine completely the state of stress at a point
2. Of the six measurements, three may be directions, so that three strains and their directions will suffice
3. All the stress may be resolved into two pure normal stresses and two shear stresses
4. The pure normal stresses will exist in directions normal to each other, and will represent the maximum and minimum values of stress existing at the point; the shear stresses will be equal and opposite and will be oriented in directions bisecting the directions of maximum and minimum normal stresses
5. Pure normal stress alone may exist at a point with no shear stress in any direction, but pure shear alone cannot exist, i.e., there must exist a normal stress in some direction, at least as large in magnitude as the maximum shear stress. In the directions of the maximum and minimum pure normal stresses (which are called the principal stresses) no shear stress exists.

So it develops that the way in which stresses may exist is somewhat limited and, as the following equation for the state of stress shows, the simplicity becomes apparent:

$$\left(\sigma_x - \frac{\sigma_p+\sigma_q}{2}\right)^2 + \tau_{xy}^2 = \left(\frac{\sigma_p-\sigma_q}{2}\right)^2 \qquad (1)$$

where σ_x = normal stress on x-plane (in x-direction); σ_y = normal stress on y-plane (in y-direction) (This does not occur in the above equation but represents a normal stress acting perpendicularly to σ_x); σ_p = maximum principal stress; σ_q = minimum principal stress; and τ_{xy} = shearing stress on x-plane (in y-direction)

The simplicity of Equation 1 becomes apparent when it is realized that only σ_x and τ_{xy} are variables at any given point. Consequently, the equation is that of a circle and defines the Mohr circle shown in *Fig.* 2. From what was said about the principal values, σ_p and σ_q must lie as shown in *Fig.* 2 (at the extremities of the horizontal diameter) and the maximum and minimum values of shear stress (τ_{max} and τ_{min}) must be the ordinates of the extremities of the vertical diameter. Now everything fits in the circle except that all the angles between the various maxima and minima are double the values that exist in fact.

There is one good explanation for this. A stress is the same whether measured in the 0-degree or 180-degree direction, just as a strain gage measures the same strain no matter which end carries the terminals. If it were possible to rotate the stress (or strain) measuring device, we would find that we were back where we started after a 180-degree rotation. Therefore, one revolution (360 degrees) around the Mohr circle must—since no point is repeated—represent 180 degrees on the specimen. Any angle on the Mohr circle measured from the direction of σ_p is 2 θ where θ is the physical angle on the stressed sur-

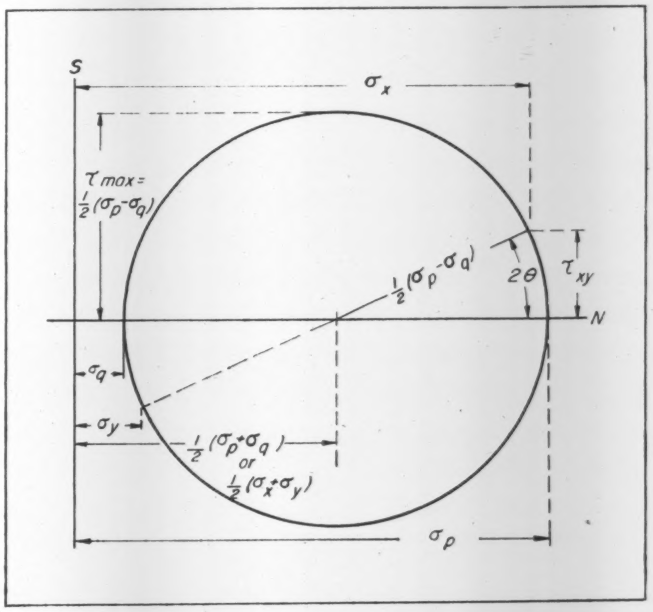

S
σ_x
$\tau_{max} = \frac{1}{2}(\sigma_p-\sigma_q)$
$\frac{1}{2}(\sigma_p-\sigma_q)$
2θ
τ_{xy}
N
σ_q
σ_y
$-\frac{1}{2}(\sigma_p+\sigma_q)$ or $\frac{1}{2}(\sigma_x+\sigma_y)$
σ_p

Fig. 2—Mohr circle of stress, indicating its relation to the principal stresses. Circle center is determined by the average of any two mutually perpendicular stresses because they lie at the extremities of a diameter

face measured from the direction of σ_p. Thus items 4 and 5 (tabulated previously) are fulfilled. To understand item 5 more fully it will be helpful to try to draw the Mohr circle for the condition of $\sigma_p = \sigma_q$. It should be noted that the stresses in any given direction are specified by the (σ, τ) co-ordinates of the corresponding point on the circumference of the circle. These are measured in terms of their projections on the N (normal) or S (shear) axis.

Also, the radius of the circle is always determined by half the difference of the principal values of the normal stress, and the center of the circle can always be found by taking the average (half the sum) of the principal stresses. In fact, and this is important, the center of the circle can always be found by taking the average of any two stresses, σ_x and σ_y, which lie at the vertical projections of the extremities of any diameter. (These are said to be orthogonal stresses, which is equivalent to saying that they are normal to each other on the specimen.) The radius is not as easily found, but it should be clear that it must bear some not-too-complicated geometrical relation to the remainder of the circle.

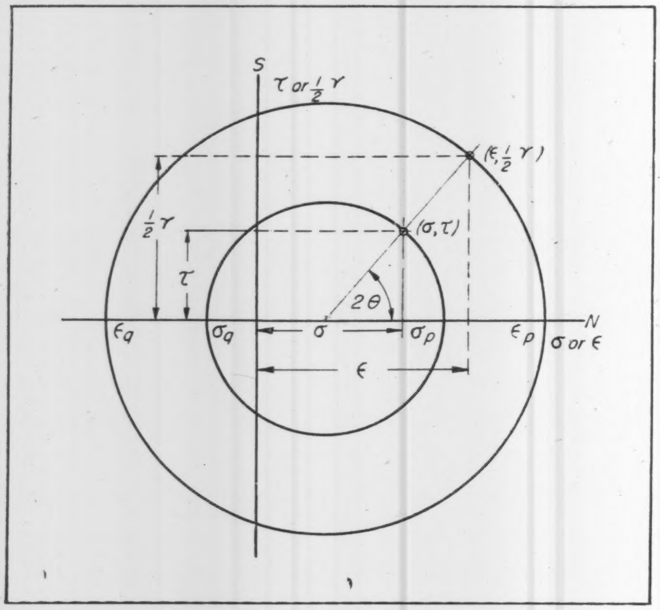


Fig. 3—Mohr circle of strain drawn concentric with the stress circle showing the determination of all shearing and normal stresses and strains at a point on a given plane by the construction of a single radius. To accomplish this the ratio of stress units to strain units must be $E_y/(1-\mu)$, whereupon the stress circle diameter is about half that of the strain circle

At this point the question may be asked: "What good does it do to comprehend the Mohr circle when we don't know the stresses, only the strains?" That is just the situation with which strain-measurers have been confronted ever since the Mohr circle was first presented to help with the understanding of the distribution of stresses for a given set of applied loads. It is something which does not lend itself too well to construction from even a set of observed normal stresses, much less strains, for after all, these stresses will be represented by the projections of their corresponding points on the circle, while we don't even know if they were projected up or down from points lying below or above the N axis. It turns out that knowing the angles between the stresses (if at least three points are known) is all that is needed, but the solution is neither straightforward nor obvious.

The Mohr Circle of Strain

The answer to this is that there is another Mohr circle—for strains rather than stresses—and, under certain conditions, it is easily constructed from three strain measurements at a point. The "certain conditions" require that the angle between each strain and the next be exactly 45 or 60 degrees (construction of both circles from three strains, 45 degrees apart in direction, will be discussed later). The 45 or 60-degree spacing explains the strain rosette gages which are available already arranged with these convenient angles between the gage directions. The equation for the strains, from which the corresponding strain circle can be produced is as follows:

$$\left(\epsilon_x - \frac{\epsilon_p+\epsilon_q}{2}\right)^2 + \left(\frac{\gamma_{xy}}{2}\right)^2 = \left(\frac{\epsilon_p-\epsilon_q}{2}\right)^2 \quad\ldots\ldots\ldots\ldots (2)$$

where ϵ_x = normal strain on x-plane (in x-direction); ϵ_y = normal strain on y-plane (in y-direction), bearing the same relation to ϵ_x as σ_y does to σ_x (see Equation 1); ϵ_p = maximum principal strain; ϵ_q = minimum principal strain; and γ_{xy} = shearing strain on x-plane (in y-direction).

Equation 2 is similar to Equation 1 except that normal strains replace normal stresses and the half-shearing strain replaces the shear stress.

The problem now reduces to this: If the three strains (with the angles between their directions as specified) are known so that the Mohr circle for strains can be drawn, how could the Mohr stress circle be determined? The answer may be grasped from the equations of the circles. It will be observed that each circle equation is of the form

$$(x-h)^2+y^2=r^2 \quad\ldots\ldots\ldots\ldots (3)$$

where h represents the circle center and r the circle radius, so that the differences between the h and r of Equation 1 and the h and r of Equation 2 must be determined. If these constants are designated with subscripts σ and ϵ as they apply to Equation 1 or 2, it can be shown that

$$\frac{h_\sigma}{h_\epsilon}=\frac{\sigma_p+\sigma_q}{\epsilon_p+\epsilon_q}=\frac{E_y}{1-\mu} \quad\ldots\ldots\ldots\ldots (4)$$

and

$$\frac{r_\sigma}{r_\epsilon}=\frac{\sigma_p-\sigma_q}{\epsilon_p-\epsilon_q}=\frac{E_y}{1+\mu} \quad\ldots\ldots\ldots\ldots (5)$$

where E_y = modulus of elasticity in tension or compression and μ = Poisson's ratio.

What all this means is that it is possible to construct both circles (stress and strain) on the same axes and with the same centers. If this is done, (make them concentric) it will be apparent from Equation 4 that the scale factor between the two diagrams is $E_y/(1-\mu)$. If the two values of h are set equal geometrically Equation 4 must have been multiplied by $(1-\mu)/E_y$. When this is applied to

Equation 5, the ratio of the two radii is $(1-\mu)/(1+\mu)$, that is, the stress circle radius is about half that of the strain circle, since Poisson's ratio, μ, is usually between one-fourth and one-third. *Fig.* 3 illustrates just how the two circles are related and how one may be constructed from the other, it being as easy to construct one from the other as vice versa.

Mohr Circle Construction from Strain Data

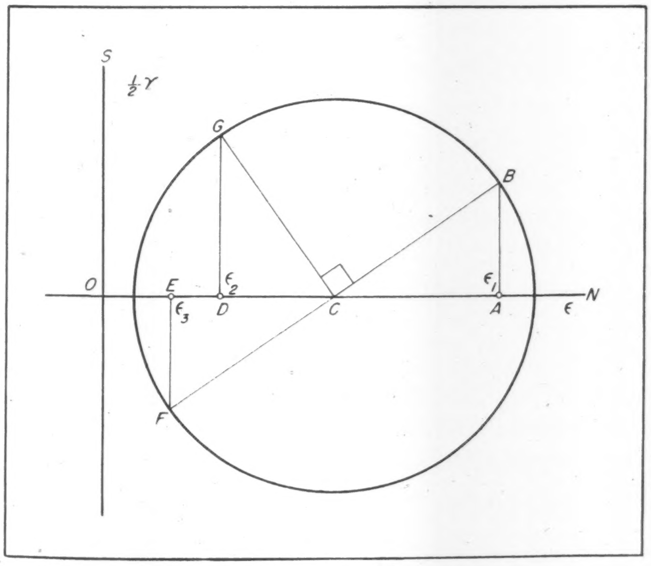
How three strains may be translated into the Mohr circles is yet to be shown. This has been done by many writers and in many ways but there is one method which is somewhat new and by its nature is easy to remember. This applies to the rectangular rosette (45 degrees between directions) and consists of locating the center of the circle and one point on the strain circle circumference whose direction is known. Then the strain circle is drawn using this point and the center, and the stress circle located inside and concentrically, with its radius $(1-\mu)/(1+\mu)$ as large. It requires the establishment of the fact that the difference of two extensional strains normal to each other determines the shearing strain along the bisector of their directions. This may be seen in *Fig.* 4 where, if *OA*, *OD* and *OE* represent ϵ_1, ϵ_2 and ϵ_3, respectively, *DG* will represent the half-shearing strain measured in a direction midway between those of the ϵ_1 and ϵ_3 strains. It is obvious that the three triangles *ABC*, *CDG* and *CEF* are congruent, so that

$$\frac{1}{2}(\epsilon_1-\epsilon_3)=\frac{1}{2}(OA-OE)=DG=\frac{1}{2}\gamma_2 \qquad (6)$$

Now everything will go along smoothly. From the three strains ϵ_1, ϵ_2 and ϵ_3, 45 degrees apart in direction (or more exactly in the directions 0 degree, 45 degrees and 90 degrees from whatever arbitrary axis is chosen), the complete state of stress may be determined in the following

1. Locate the center of the circle, $h=\frac{1}{2}(\epsilon_1+\epsilon_3)$
2. Erect a vertical line at ϵ_2
3. Construct the horizontal line where ordinate is $\frac{1}{2}\gamma_2=\frac{1}{2}(\epsilon_1-\epsilon_3)$
4. Find the intersection of the horizontal and vertical lines just constructed and draw a circle through this point $(\epsilon_2, \frac{1}{2}\gamma_2)$ with the center at *h*. This is the Mohr circle of *strain*
5. Draw another circle with the same center but with a smaller radius $(1-\mu)/(1+\mu)$ as long. This is the Mohr circle of *stress*. Whatever scale was used for strain must be multiplied by $E_y/(1-\mu)$ to find the stresses
6. Draw a diameter normal to the radius through $(\epsilon_2, \frac{1}{2}\gamma_2)$ and label the ends 1 and 3 remembering that the point 2 on the circles represents measurements in the 45-degree direction, while points 1 and 3 represent measurements in the 0 and 90-degree directions, paying attention to the sign of the rotation (counter clockwise usually being taken as positive). The angle which the direction of principal stress makes with the 1 or 0-degree direction is one-half that measured from the direction of the point 1 to that of the *N* axis.

Fig. 4—Relationships existing between two mutually perpendicular normal strains and the half-shearing strain on the plane oriented midway between their directions



Now all the stresses, strains and directions for the point in question have been determined. The values of principal stresses may be measured easily at the circle extremities and the angle from their direction to the 0-degree direction easily measured. Even if all these values are calculated it will still be helpful to make a rough sketch of the two circles just to make sure that no obvious mistake has been made. After all, when the graphical solution is once established and the projections from the points 1, 2, and 3 fall upon the values ϵ_1, ϵ_2 and ϵ_3 it is hardly possible that any mistake has been made, for both signs and magnitudes have been checked. Even when only one or two strains have been measured at a point it is usually desirable to consider them in the light of what Mohr circle solution may exist, making whatever assumptions (such as knowing the direction of principal stresses) are necessary to fill out the rest of the data.

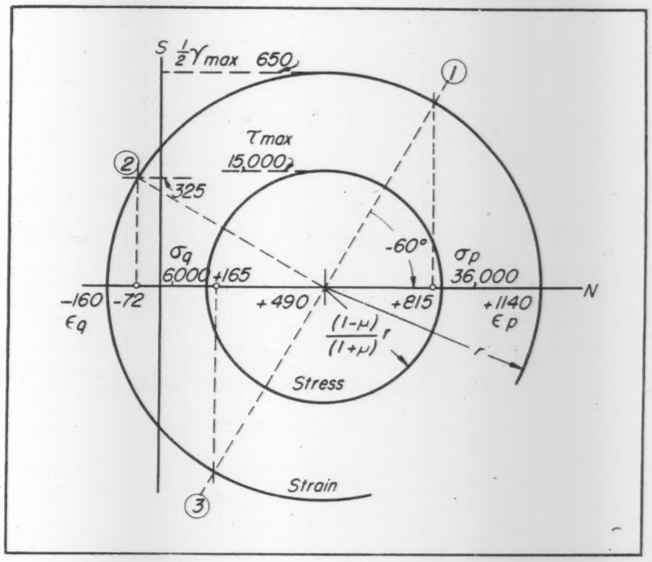


Fig. 5—Numerical example showing the complete state of stress solved from three measured strains at one point

A numerical example is solved graphically in *Fig.* 5. The given data are ϵ_1 = +815 microinches/inch at 0 degree;

$\epsilon_2 = -72$ microinches/inch at 45 degrees; $\epsilon_3 = +165$ microinches/inch at 90 degrees; $\mu = 0.3$ (Poisson's ratio); and $E_y = 30 \times 10^6$ psi (Young's Modulus)

The solution shows $\sigma_p = +36{,}000$ psi; $\sigma_q = +6{,}000$ psi; $\tau_{max} = 15{,}000$ psi; and $2\,\theta_p = -60$ degrees ($\theta_p = -30$ degrees)

The angle between the maximum principal stress σ_p, and σ_1 the stress in the 1 direction, is given as negative since the direction of 0 degree was arbitrarily assigned to the first gage, with angle increasing counterclockwise through the remaining gages. From this figure it may readily be shown that

$$\theta_p = 45^\circ - \frac{1}{2}\ arctan\ \left(\frac{\epsilon_1 - \epsilon_3}{2\epsilon_2 - \epsilon_1 - \epsilon_3}\right)$$

This example (and one or two more with arbitrary strain data) should be solved to get the relation of the two circles firmly in mind, even if a rosette analysis is never made. It should be noted that the relation of stress to strain is generally not E_y, that the sign of the stress is not always the same as that of the strain, and particularly that the maximum stress may differ greatly from the largest apparent stress (strain × modulus). In the numerical example this difference is about 50 per cent.

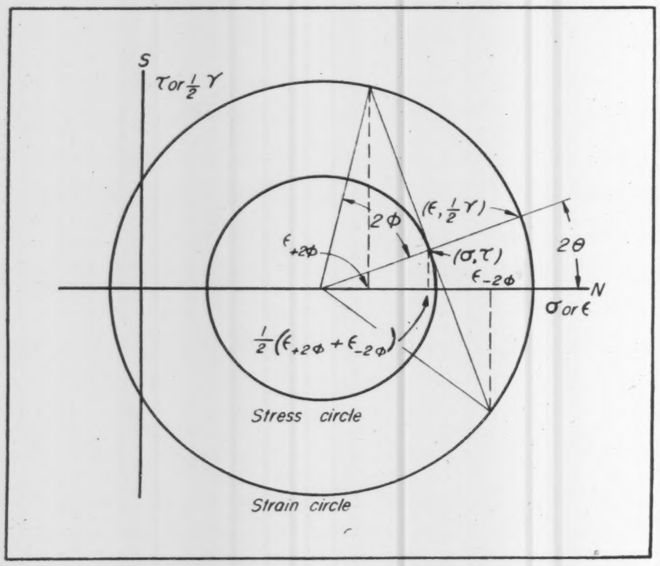


Fig. 6—Circles for stress and strain showing how a V-type stress gage averages two strains to yield an output proportional to the normal stress existing in the direction of its axis. The V-configuration is not essential to stress gage operation but is perhaps the most convenient besides being most easily proved by graphical means

The Stress Gage

Often, since the introduction of stress gages in 1943, there has been raised the question, "Since stresses are associated with internal forces and their only external manifestations (so far as the SR-4 bonded-wire unit is concerned) are strains, how can stresses be measured direct by such a means?" Strictly speaking, they cannot, but to the extent that stresses can be calculated from strains, they can. The calculation can be accomplished readily. To understand this it is best to return to the Mohr circles (of stress and strain) even though no attempt is being made to determine with one stress gage any more than the stress in a given direction—as contrasted to the complete solution available from a rosette analysis.

Considering *Fig.* 6, our concern is with the measurement of stress in the direction θ. It is no problem to measure the strain, ϵ, with a strain gage. Measurement of the stress, σ, is a little more obscure. We realize that, if it were possible to measure a strain which corresponded to the projection of the point (σ, τ) on the strain axis, it would be necessary only to multiply by the scale factor $E/(1-\mu)$ to determine the stress. A little thought will show how this may be done.

If a line is drawn tangent to the stress circle at the point (σ, τ) so that it intersects the strain circle at the points $\epsilon_{+2}\phi$ and $\epsilon_{-2}\phi$ it will be appreciated that the average of these two values, $\frac{1}{2}(\epsilon_{+2}\phi + \epsilon_{-2}\phi)$ will always be proportional to the quantity sought. This is the basis of the stress gage. Two equal components, each making the proper angle with that of the desired stress, are averaged to produce a quantity proportional to that stress. This is most easily accomplished by making the path of the wire filament that of a Vee, the included angle being determined by the geometry of the Mohr circles of stress and strain. Since the angles on the Mohr diagrams are actually twice those existing physically, the total included angle of the Vee (2ϕ) will be just equal to the angle between the radius through the point (σ, τ) and either of the radii through the points corresponding to the projections $\epsilon_{+2}\phi$ and $\epsilon_{-2}\phi$. The included angle is termed 2ϕ because the apparent strain resulting from the stress gage may be represented as follows:

$$\frac{1}{2}(\epsilon_{+2\phi} + \epsilon_{-2\phi}) = (\epsilon_x + \epsilon_y\ tan^2\ \phi)\ cos^2\phi = \frac{\sigma_x}{\left(\frac{E_x}{1-\mu}\right)} \quad \ldots\ldots(7)$$

This equation agrees both with the fact that the apparent strain thus measured must equal the stress divided by the scale factor, $E_y/(1-\mu)$, and with the elementary biaxial stress equation that would have been used to solve for the stress in one direction from the readings of two strain gages mounted at right angles:

$$\sigma_x = \frac{E_y}{1-\mu^2}(\epsilon_x + \mu\epsilon_y) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

provided that $tan^2\ \phi = \mu$. That this latter is true obtains from the relation between the radii of the two Mohr circles, which yields $cos\ 2\phi = (1-\mu)/(1+\mu)$ as can be seen in *Fig.* 6. When this is applied to the identity $tan^2\ \frac{1}{2}(2\phi) = (1 - cos\ 2\phi)/(1 + cos\ 2\phi)$, $tan^2\ \phi = \mu$.

All this may seem to be the long way around to designing a stress gage, but the terminology was developed to express the effective Vee angle for any gage possessing a lateral (transverse) sensitivity. Thus, when it is stated that a gage has a lateral sensitivity of 0.02, it is equivalent to stating that it is effectively a Vee gage of total included angle 2ϕ, such that $tan^2\ \phi = 0.02$, and $2\phi \approx 16$ degrees (approximately). Actually, this value of 16 degrees has little significance in strain gage manufacture for the apex of the Vee cannot be made infinitely sharp, so that there

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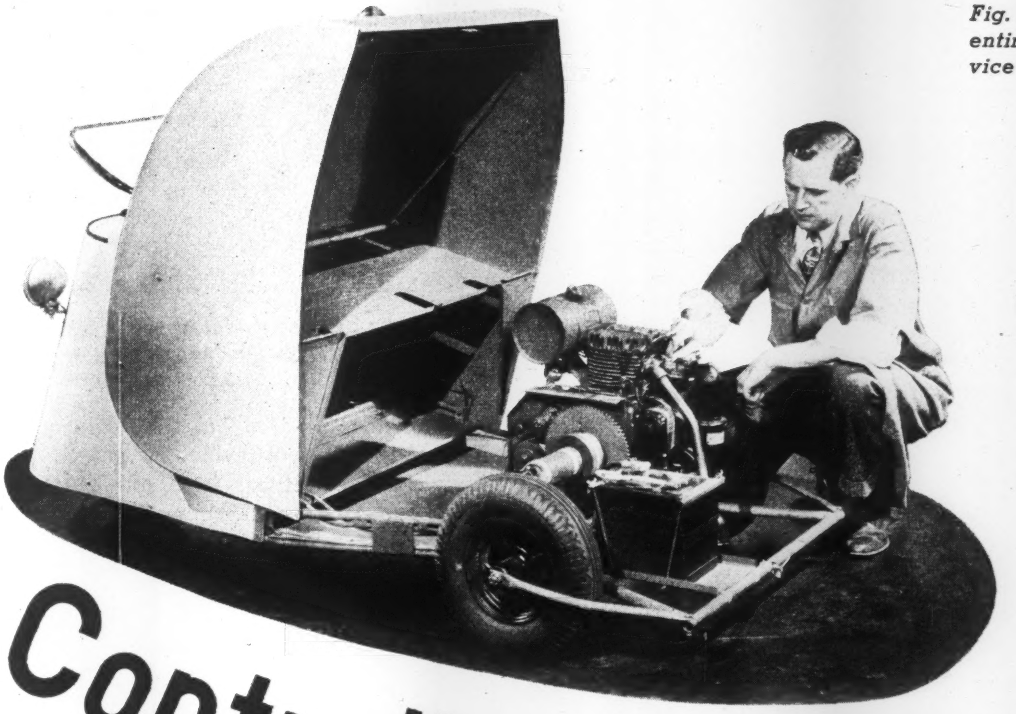

Fig. 1—Mercury clutch provides entire automatic transmission device for this light-delivery vehicle

Controlled Clutching *Licks* HIGH BREAKAWAY TORQUES

By H. J. Zimmermann
Chief Engineer
Mercury Clutch Corp.
Canton, Ohio

SINCE most of our useful prime movers—electric motors, internal combustion engines, turbines, etc., develop maximum torque at relatively high speeds, it usually becomes necessary to interpose a clutch between the driving and driven elements. Thus, under "no load" conditions, the prime mover can attain the speed at which it develops sufficient torque to overcome the resistance of the driven load. This is especially desirable and oftentimes vital where a direct-connected load offers resistance to such an extent that the starting torque capacity of the most economical size of prime mover is far exceeded.

Manual clutch arrangements are not always satisfactory—some machines would have output rates severely limited, others would be subject to early deterioration from premature engagement, still others would be too clumsy and some not at all practical. Too, the complexity of some mechanically-actuated automatic-clutching arrangements eliminates incorporation of such a design.

Centrifugally-controlled clutches, tailored exactly to machine requirements as in *Fig.* 1, offer much toward achieving the most economical design whether it be a vehicle or possibly an automatic washing machine. Such a clutch is usually a friction-shoe type, controlled and engaged by the action of centrifugal forces. Centrifugal force actuation may be obtained in two general ways—mechanically or hydraulically. Since mechanical type flyweight-actuated centrifugal clutches are well known, this article will dwell primarily upon the hydraulic design.

Performance of any type of centrifugal clutch can be controlled to provide engagement at specific speeds or speed intervals. Engagement can be retarded by springs designed to delay centrifugal action until the specified speed—thus centrifugal force—is reached.

Predetermined delayed action is usually desirable in

UTILIZATION of centrifugal force with mercury as a hydraulic medium affords a measure of clutching control and automaticity not heretofore realized. The inventor here discusses the basic design features of such clutches and outlines a few of the advantages and applications

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"precise control" clutches, especially where no engagement should occur within a period of low-torque speed. Without this feature, actual engagement would begin almost instantaneously, and the centrifugal force which increases progressively as the square of the speed, would reach considerable magnitude at low speeds.

If the centrifugal force acting on a rotating body is 100 pounds at a certain speed, it is 2^2 times or 400 pounds if the speed is doubled, and 3^2 times or 900 pounds if speed is trebled. A mass *m* weighing one pound when rotating on a one-foot radius will produce a centrifugal force of

$$F = mR\omega^2 = \frac{1}{32.16} \times 1 \times \left(\frac{\pi N}{30}\right)^2 = 0.000341N^2 \quad \ldots\ldots\ldots(1)$$

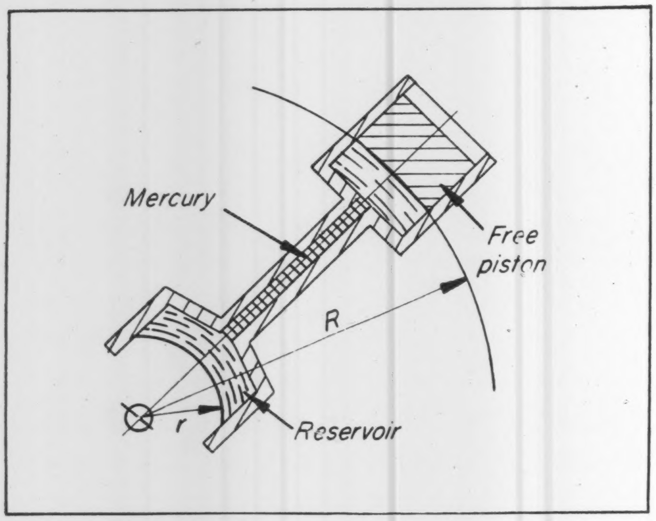


Fig. 2—Above—Simplified sketch showing centrifugal actuation of free piston by means of mercury head

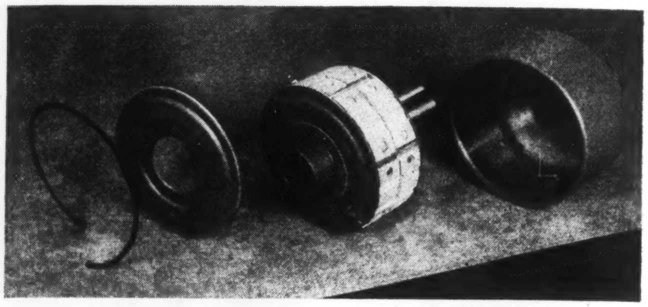


Fig. 4—Disassembled production model of radial-pressure clutch showing friction segments

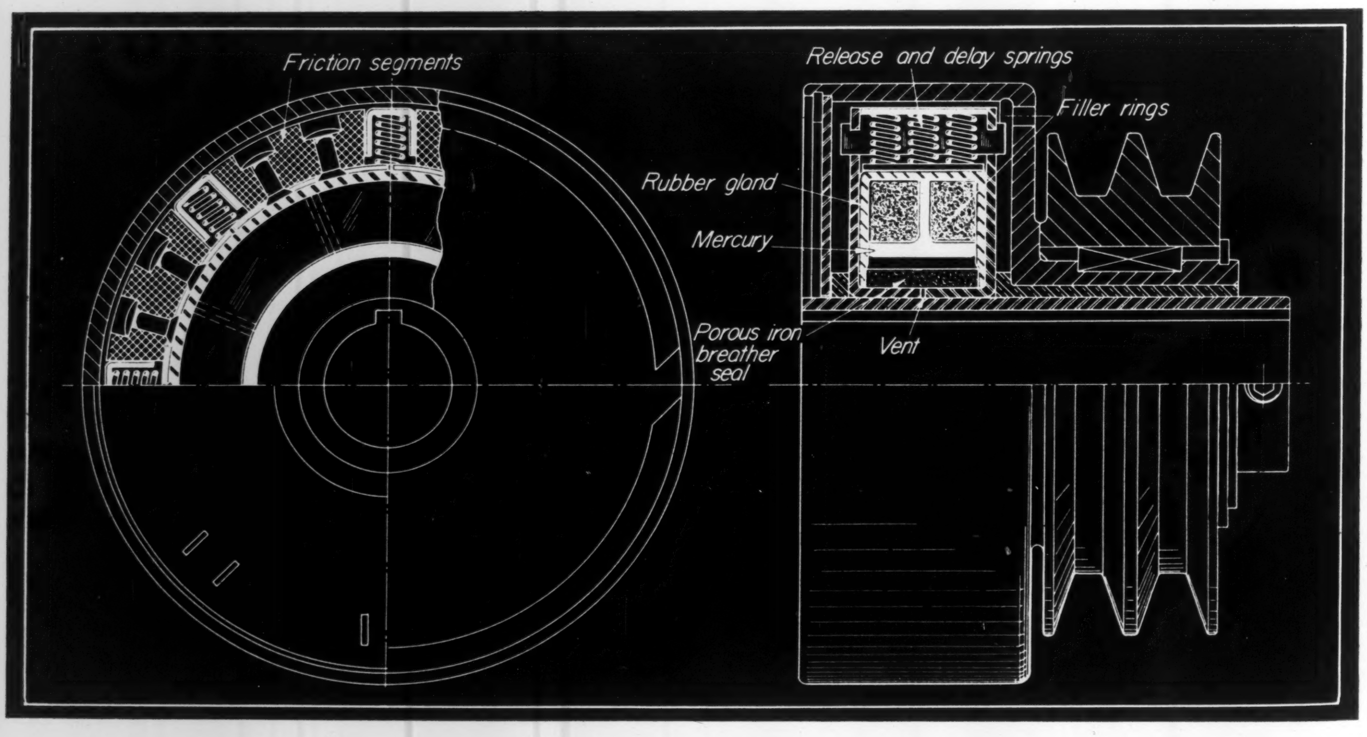
where R = the radius of gyration in feet, ω = the angular velocity, and N = the number of revolutions per minute. Thus, at 1000 rpm the centrifugal force of the pound weight will be 341 pounds and at 2000 rpm it will be 1364 pounds.

In considering the design of a centrifugal clutch, this mass effect is important. Consequently, to assure a minimum of volume and a maximum of active weight as well as the desirable characteristics of a hydraulic fluid, mercury is used to provide the mass for centrifugal clutch actuation. As a simplified example of this principle, consider centrifugal forces as acting on a column of mercury and this total force created as acting upon an enlarged area somewhat similar to a free piston. Thus, as shown in *Fig.* 2, pressure *P* produced by such a piston arrangement with a radial reservoir is:

$$P = \frac{\delta\left(\frac{\pi}{30}\right)N^2}{24g}(R^2 - r^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

With speed N = 1000 rpm, density of mercury δ = 0.4902 lb/in³, R = 12 inches, and r = 1 inch, the pres-

Fig. 3—Below—General cross-sectional details of a radial-pressure mercury-actuated clutch



sure developed by the mercury head becomes

$$P=6.964(R^2-r^2)=6.94\times143=995.9 \text{ psi}$$

Thus, assuming approximately 1000 psi, the pressure exerted on a 3-inch piston is roughly 7100 pounds. This force is independent of the diameter of the tube connecting the reservoir and piston chamber and merely depends upon the height of the column $(R - r)$, or head. Thus only a small volume of mercury is necessary.

In actual clutch construction, rather than the tube of mercury and piston shown in *Fig.* 2, an annular rubber gland is used. A clutch of this type is shown in *Fig.* 3. Centrifugal forces through the mercury medium act upon the rubber wall and thus against the friction segments. The springs counteract the mercury pressure and can be selected so that "delayed" engagement occurs at any desired speed. To utilize the fact that only the head of mercury is concerned, volume is reduced to a minimum by means of the filler ring shown. The mercury which forms the centrifugal head is confined to a number of small radial holes in this filler ring.

Mercury, which lies in the bottom of the gland when the clutch is at rest, is taken along by friction until a uniformly distributed annular ring is formed during acceleration of the rotor. When the rotor develops sufficient speed so that the centrifugal force of the mercury, aided by that of the brake shoes, overcomes the set resistance of the springs, the shoes begin to move into clutching position. So long as the motor maintains full speed, maximum clutching action is present, but on severe speed reduction the driven load is automatically released. A clutch of this type is shown in *Fig.* 4.

A bushing of porous iron at the inner diameter of the

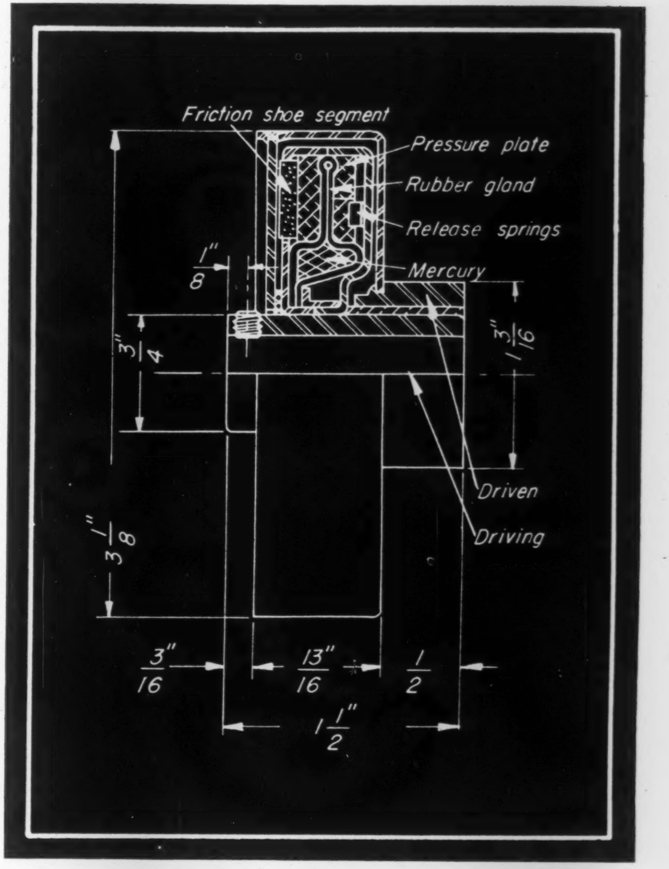
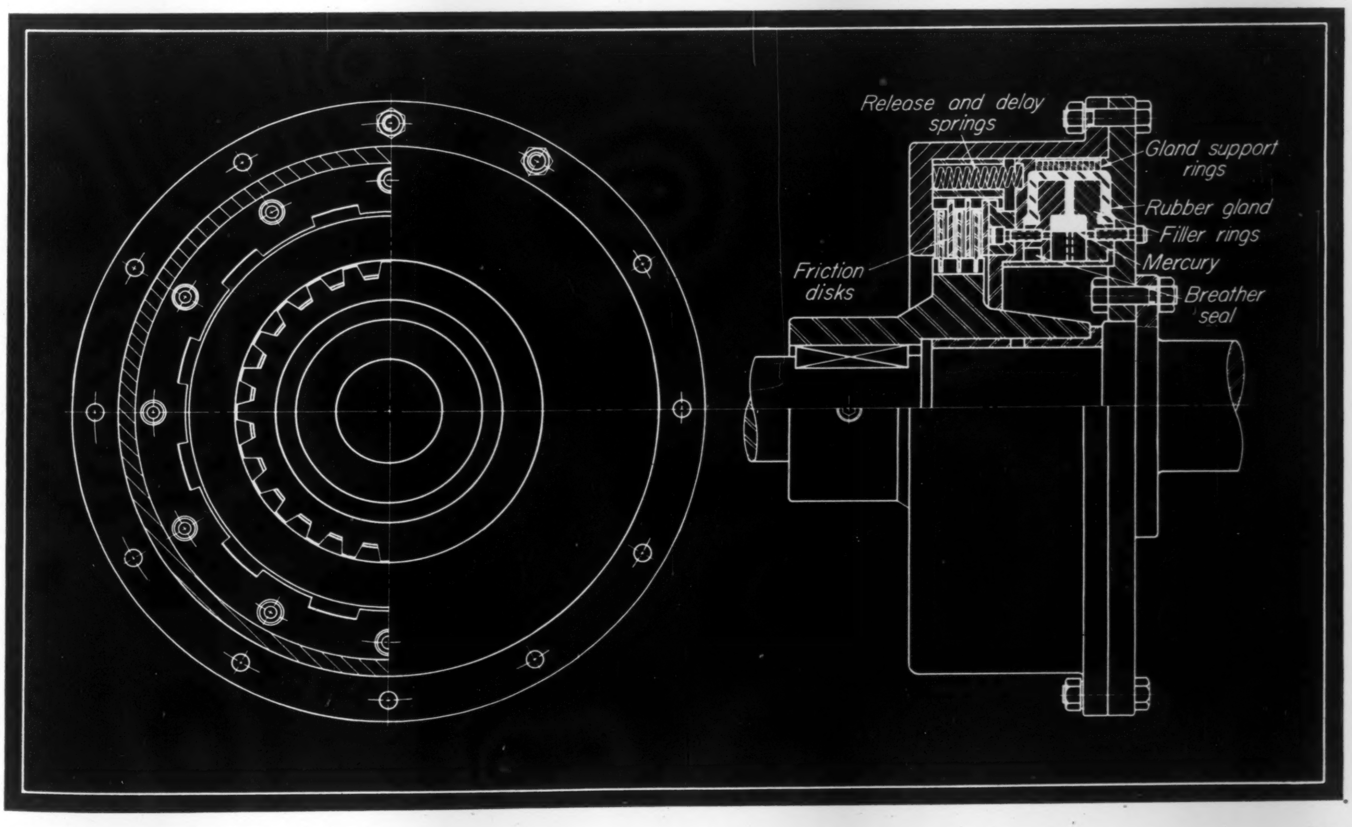

Friction shoe segment
Pressure plate
Rubber gland
Release springs
Mercury
1/8"
3/4"
1 3/16"
3 1/8"
Driven
Driving
3/16"
13/16"
1/2
1 1/2"

Fig. 5—Above—Cross-sectional details of small-size axial-pressure mercury-actuated clutch

Fig. 6—Below—Details of large-size axial-pressure clutch with multiple friction disks to suit the required capacity


Release and delay springs
Gland support rings
Rubber gland
Filler rings
Mercury
Breather seal
Friction disks

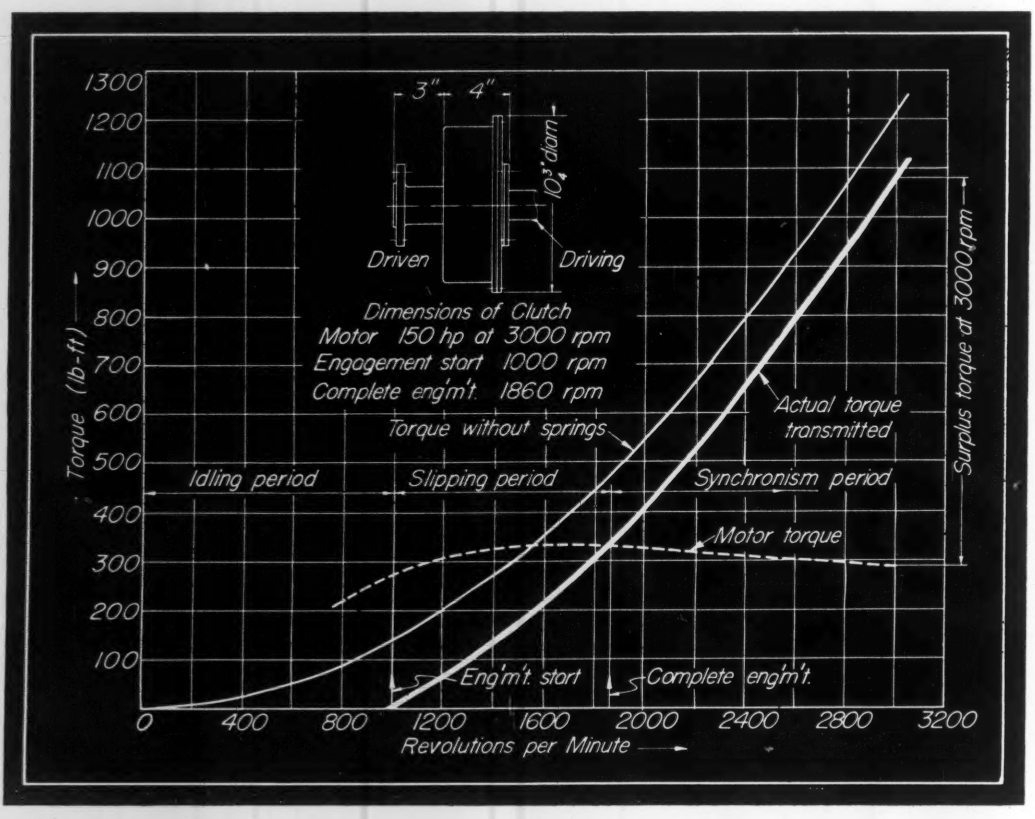


Fig. 7—Left—Performance curves of clutch similar to that shown in Fig. 6 used on a recent helicopter installation

gland, *Fig.* 3, serves as a seal and also as a breather to avoid creating a vacuum or pressure in the gland.

Action of mercury, utilized to produce radial pressure in the clutch shown in *Fig.* 3, can also be used axially as illustrated in *Fig.* 5. Here, since fluid pressure created is always exerted in all directions, motion or work is produced only in the direction the confining walls permit. Actuating pressure created, of course, is approximately zero at the inner surface and reaches a maximum at the periphery of the gland.

Axial Clutch More Universal

This axial type clutch permits the use of a greater number of helical springs which afford a more favorable spring rate and facilitate adjustment of the engagement speed. Where a larger number of friction disks are necessary with a correspondingly greater stroke, a cylindrical gland sustained radially by a series of thin rings to allow for axial expansion is utilized, *Fig.* 6. A filler ring again reduces the amount of mercury required.

The axial clutch design is by far the most versatile—capacity can be set as desired by merely using the proper number of friction disks and varying the height of the mercury column. Design similar to that of *Fig.* 6 is usually used for capacities upwards of 100 horsepower. However, these clutches can be designed for 1000 horsepower or more and have been used successfully for air compressor drives, portable "buzz" saw drives, and many other industrial applications. Time delay due to inertia of mercury makes the mercury-actuated clutch especially suitable for drives similar to that of the electric motor-driven compressor. Capacity of the clutch must be of such magnitude that it will overcome the breakaway torque of the compressor—sometimes as much as three times the running torque. The motor can attain full speed before the clutch engages whereupon the torque of the motor and the inertia of the armature are sufficient to start the compressor.

Ability to accelerate slowly any driven element having considerable inertia is vitally necessary on such machines as helicopters, centrifuges, etc. Small helicopters present little problem but as designs increase in size, proper clutching prevents severe loadings of the blade mechanisms and transmission as well as protects the structure from sudden torque fluctuations. Delay springs prevent transmission of power to the blades at idling speeds and engines can be "preflighted" without a manual control.

Performance of a helicopter clutch for a 150 horsepower, 3000 rpm engine is shown in *Fig.* 7. Release springs are selected so that the clutch engages or releases at 1000 rpm. Torque reduction effected by the springs is constant as indicated in the curves. Engagement occurs at the intersection of the clutch curve with the engine-torque curve and from that point on there is a surplus of clutch torque. This surplus, at a running speed of 3000 rpm, is about 2.7 times that of the running torque which constitutes an adequate factor of safety.

Illustrative of the axial designs utilized for capacities under 100 horsepower is that shown in *Fig.* 5. This design for a ¼-horsepower, 1750 rpm electric motor is utilized for automatic washing machines. Friction shoes for this design are segments held in openings in the side plates and delayed engagement is accomplished by means of flat arch springs arranged between the side and pressure plates.

Performance data for this clutch is shown in *Fig.* 8. Curves show comparative current consumption of the short-hour, split-phase motor with and without the clutch. In starting the load without the mercury-actuated clutch, potential drops from 110 volts to 90 volts—a common occurrence when household circuit demands exceed 20 am-

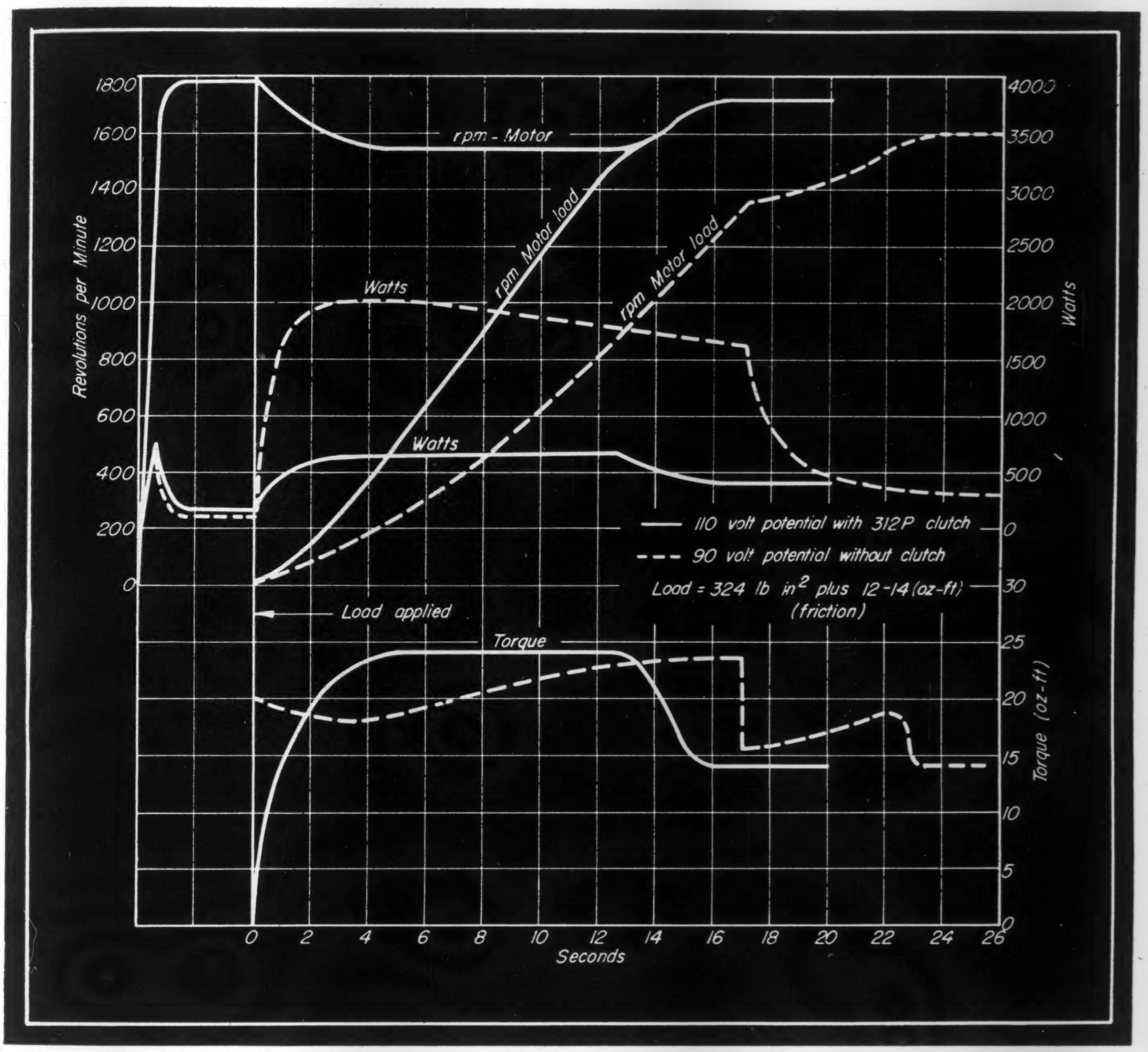
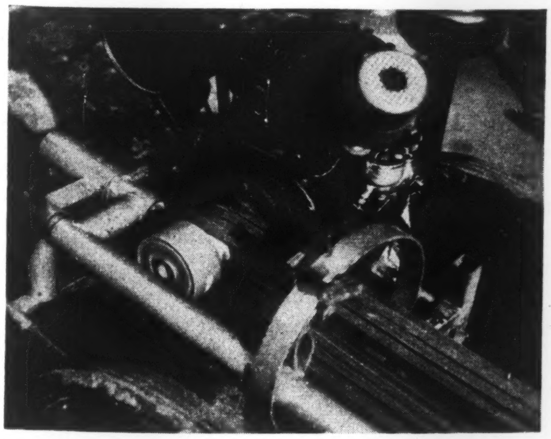


Fig. 8—Performance curves for a short-hour, split-phase, ¼-hp, 115-volt, 1750 rpm, 60-cycle electric washing machine motor with and without a mercury-actuated clutch

Fig. 9—Clutch in Fig. 4 provides drive for saw driven by a 6-hp, 2200 rpm gasoline engine



peres. Current consumption rises to 1970 watts and, rated capacity being a ¼-horsepower input of 366 watts, overload was 540 per cent.

With the clutch, the motor reaches full speed within about ½-second, and at the point of engagement speed drops to 1550 rpm. Maximum current consumption is only 625 watts indicating an overload of 1.7 times the input watts at rated load. Under actual conditions acceleration time of 24 seconds without the clutch is reduced to 16 with the clutch.

Since most electric motors accelerate and decelerate rapidly, time delay caused by the inherent inertia of mercury is found sufficient. Thus delayed-action springs or release springs would not be necessary other than for positive release of the friction segments. They are necessary, however, where a specific clutching speed is desired, such as with internal-combustion engines, to allow for an idling period and with other special applications, *Fig.* 9. The mercury-actuated clutch also can be designed so that it disengages at a specific predetermined high speed. In addition to the foregoing, the centrifugal force of mercury can be utilized to actuate braking devices for preventing overspeeding, safety devices for the operation of hoists or elevators, and special-purpose couplings.

Evaluatin

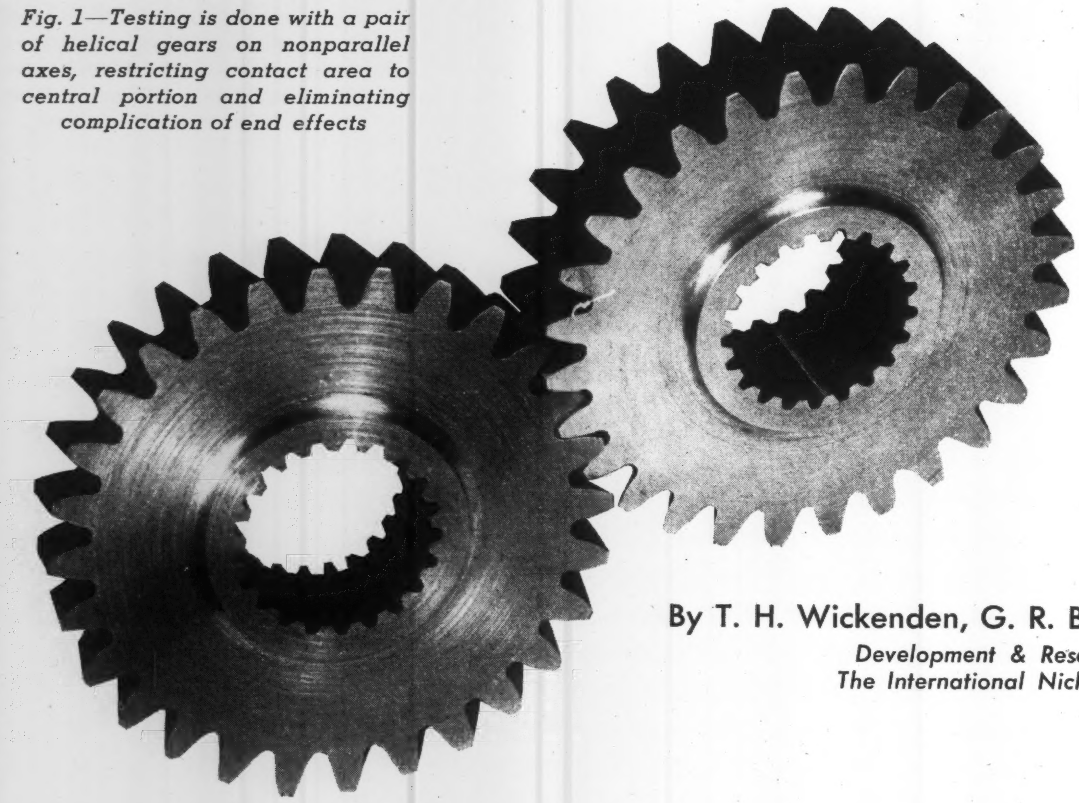


Fig. 1—Testing is done with a pair of helical gears on nonparallel axes, restricting contact area to central portion and eliminating complication of end effects

By T. H. Wickenden, G. R. Brophy & A. J. Miller

Development & Research Div.
The International Nickel Co. Inc.

VARIOUS methods for evaluating the surface durability of gears, which include metallurgical tests, tests of gears, and wear tests of specimens designed to simulate gear conditions, have not always indicated service experience accurately. Recognition of the need for an improved technique has resulted in an attempt, which will be described in this article, to develop a suitable machine and method of test. The aim has been to test gears under controlled conditions of load, temperature, speed and lubrication, and to eliminate, as far as possible, the unavoidable, complicating variables encountered in the manufacture of gears. Primary concern in these tests is with the deterioration of the teeth surfaces which fail by pitting rather than with the beam fatigue strength of the teeth. Ultimately, it is hoped, a comparison of gear materials and their preparation may be made.

This article covers a description of the testing machine in its finally developed form, the test specimen, and the test procedure. One of the important problems was the determination of the point of failure of gear surfaces, the solution of which is explained in detail. A few test data are presented to demonstrate the results obtainable. Specimens and the machine were designed by J. O. Almen, J. C. Straub and F. J. Johnson of the General Motors research laboratory in cooperation with T. H. Wickenden and E. J. Hergenroether of the International Nickel Co.

Shown in *Fig.* 1, the test specimens consist of a pair of 28-tooth helical gears with pitch diameter 3.5302 inches; normal pressure angle 20 degrees, helix angle 7½ degrees and tooth face width 1 inch. In operation, the two gears are opposed and rotate in different planes which include an angle of 15 degrees, thereby developing an elongated elliptical contact area which is more or less restricted to the central portion of the tooth, eliminating the complication of end effects. The area of contact may be visualized by considering the contact between two crossed cylindrical surfaces. By thus restricting the contact area, the test stresses may be increased to the desired maximum without approaching the beam endurance limit of the teeth. With the particular test gear used, the upper limit of stress is about 250,000 psi.

AN INGENIOUS TESTING MACHINE for determining the surface durability of gears is discussed in this article, which is based on a paper presented at the recent annual meeting of the American Gear Manufacturers Association in Hot Springs, Va. Typical test data are included and an account is given of the testing procedure, which employs a noise meter to determine the point at which destructive failure begins

The machine, *Figs.* 2 and 4, operates on the "four-square", or closed circuit, principle in which two pairs of gears are opposed—one pair a permanent part of the machine and the second pair the test specimens. The usual design was modified to accommodate opposed gears rotating in different planes, by arranging the shafts at an angle of 15 degrees and the insertion of four constant-velocity universal joints in the shaft lines to bring the two gearboxes to the same level. Two universals were used on each shaft line to reduce the operating angles of each to safe values. Details of construction of one of the uni-

urface Durability of Gears

versal joints and one of the gearboxes with gears, bearings and seals in position are shown in *Fig.* 3. The other universals and gearbox are similarly constructed.

Another feature of the modified design is the mounting of the test gearbox on trunnions so that a torque may be applied to the system by means of weights placed on a pan hung from a torque arm, effectively 12 inches long, which in turn is fastened to the box. The reaction to this torque develops the stress in the gear teeth.

Each gearbox is equipped with a transparent cover and is kept filled to capacity with oil which is pumped during operation in a continuous flow through the box under a constant pressure of 30 psi. The oil is supplied through flexible lines and valves shown in *Fig.* 4 from a 5-gallon tank, and is maintained at any desired temperature to −5F by means of a thermostatically controlled immersion heater and a water-cooled radiator. The circulating pump, radiator and oil filter are mounted above the tank and may be seen in *Fig.* 2.

The gear train is driven by three V-belts at a speed of 900 rpm by a 3-hp, 1200-rpm induction motor mounted beneath the table at an angle of 7½ degress from the horizontal. In the even of an unexpected failure of any of the rotating parts the increased power operates an adjustable overload relay and a magnetic brake, visible in *Fig.* 4,

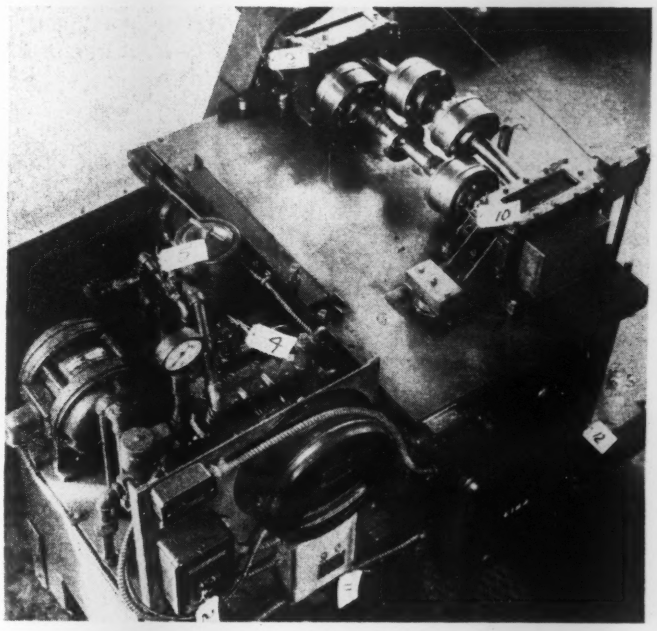
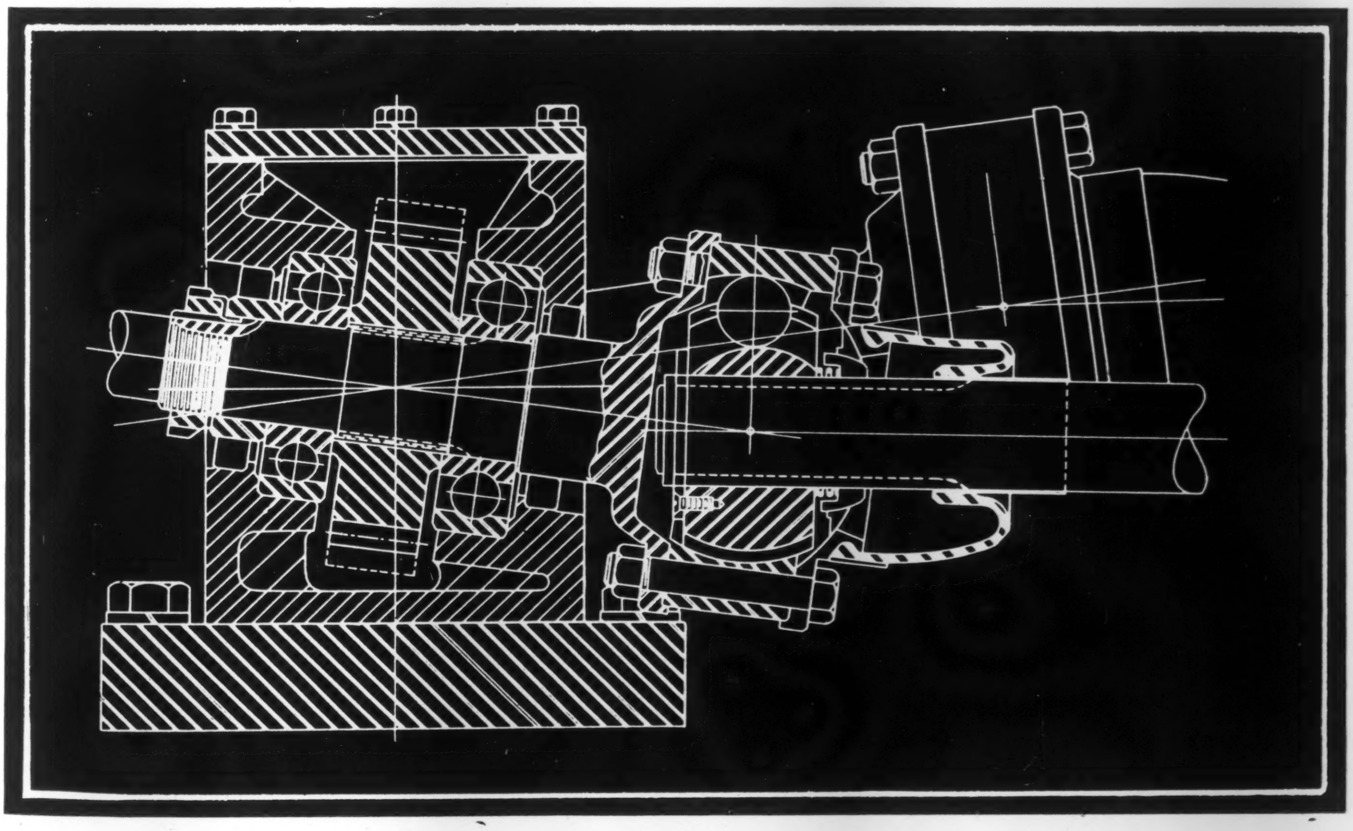


Fig. 2—Above—Top view of gear-testing machine, showing oil pump system and temperature controls

Fig. 3—Below—Details of construction of universal joint, gearbox, gear, bearings and seals



to stop the machine. In addition there is an adjustable limit switch placed immediately beneath the torque arm, which is actuated in cases of excessive wear, tooth fracture or serious vibration.

In starting a test, the oil temperature control system is placed in operation. When the desired temperature is reached the load is placed on the pan and supported, the motor is started, the load released and the reading taken on a Veeder counter direct-connected to the driven pulley, *Fig.* 4. Oscillatory vibration of the system is prevented, except at the very lightest loads, by a dash pot placed under the counter balance arm at the rear of the test gearbox.

Noise Meter Aids Test

With the idea that, as the teeth surfaces deteriorated under test conditions, the power losses would increase, a recording power meter was connected to the motor power line. The power record, together with visual examination, demonstrated that the gear efficiency continued to improve even after pitting was well advanced. On the other hand the deterioration of any of the rotating parts of the system—joints and bearings—was recorded. In a few early tests this gave false indications of end points. Because of its unreliability this method of indication was discarded. It was noticed, however, that when the gears had pitted seriously, the volume and tone of the sound emitted from the machine had changed audibly. This suggested that a noise meter might yield a dependable indication of destructive pitting.

Accordingly, a simple and inexpensive device[*] was assembled which measures the noise levels in terms of millivolts. It consists of a "Kontak", high impedance microphone, mounted on the weight lever, feeding into an amplifier and vacuum tube voltmeter. A continuous record of the voltage output and, therefore, noise level is obtained with a potentiometer recorder. The wiring diagram of the device in its final form is shown in *Fig.* 5.

A few typical examples of noise level records, replotted on a logarithmic time scale to sharpen the end points, are shown in *Fig.* 6. No real significance should be attached to the initial millivolt level of any tests, for that may be adjusted at will, and has been, in order to space the records. Failure or destructive pitting of the teeth surfaces is marked by a continued rise of the noise level.

Among numerous tests already conducted were some on gears made from SAE 4645 steel. Forged blanks were annealed, turned, and splines and teeth roughly cut. The rough gears were then oil quenched and tempered to three levels of hardness: Rockwell C 30, 36 and 43, after which they were finished on a Fellows gear shaper. Because of the procedure of finish machining after heat treatment, the hardness of the test gears was limited to rockwell C 45.

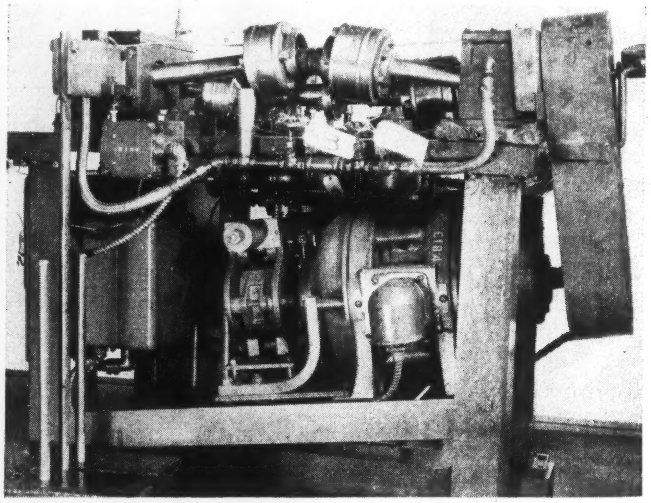


Fig. 4—Above—Side view of gear-tester, showing drive shafts, motor, brake, and weight for load application

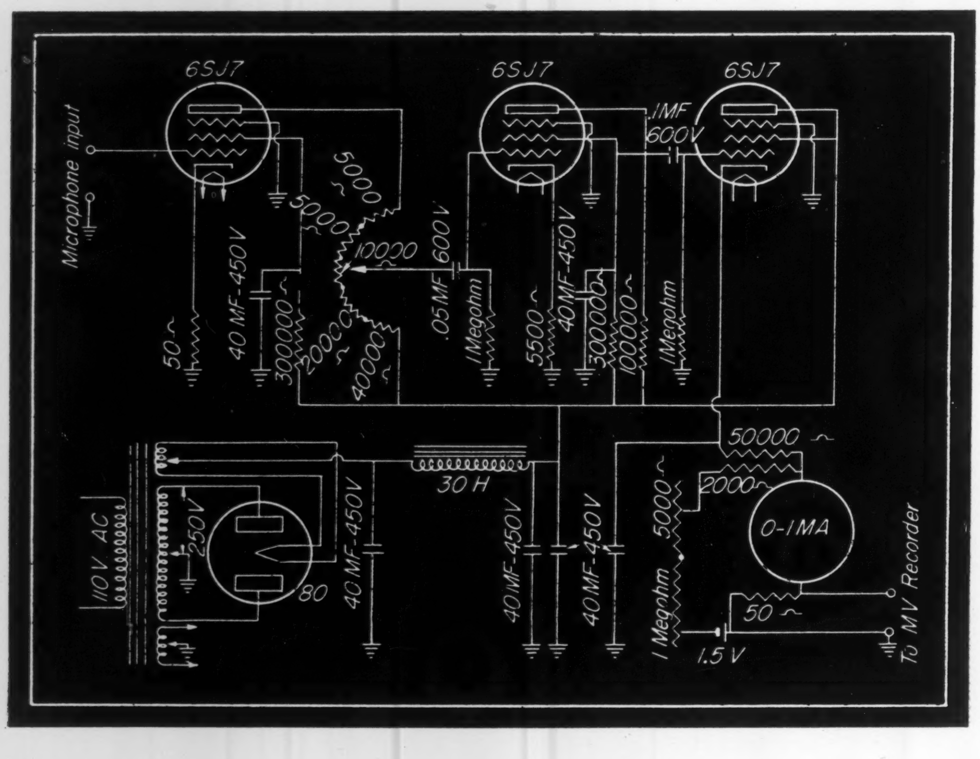


Fig. 5 — Left—Circuit diagram for noise meter designed for use in conjunction with gear-testing machine

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[*] Originally conceived, designed and constructed by A. P. Edson, formerly of the International Nickel Company's Research Laboratory, and improved to its present form by A. J. Miller.

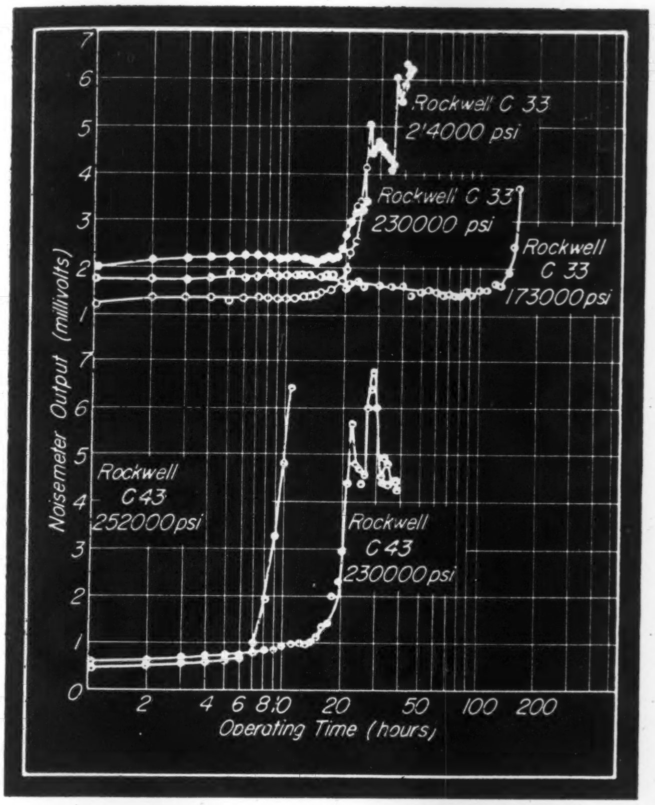


Fig. 6—Typical gear test noise records. Failure or destructive pitting is marked by continued rise in noise level

All tests were conducted at 215 F in the manner of fatigue tests in order to develop the stress-life relation and to establish, if possible, the stress below which the gear life is indefinitely long. Stress-life curves at each level of hardness are shown in *Fig.* 7. These have the shape characteristics of fatigue curves. When the indicated stresses for lives of 5 × 10^7 cycles are plotted against the hardnesses of each group, the curve of *Fig.* 8 results, which again is characteristic of fatigue in that higher stresses for indefinite lives are found as the hardnesses increase. It is fully expected that this trend will continue to higher hardnesses than those studied. Indeed, the performance of the permanent gears of the machine furnish excellent support for this expectation. They are made of carburized 4600 steel, hardened and tempered to rockwell C 63, and have withstood, without distress, the summation of all tests conducted to date—a total of more than five hundred million cycles of calculated stresses ranging between 150,000 and 250,000 psi.

Calculation for the compression stress in the teeth is made through the application of the following equation[†]

$$s_c = \frac{23,600}{\alpha \times \beta} \sqrt[3]{\frac{{}_4T(\tan^2\alpha_n \cos^2\Delta)^2}{\cos\Delta \ \sin^2\alpha_n}}$$

in which α_n = normal pressure angle = 20 degrees, Δ = helix angle = 7.5 degrees, α and β are functions of the involute tooth form and the angle between the contact line (comparative length and breadth of the contact area), $\alpha \times \beta$ = function of the contact area, and T = torque in inch-pounds on the gear = ½ loading torque.

Simplified, the foregoing equation becomes

$$s_c = 15,600\sqrt[3]{T}$$

The formula is rigorously mathematical. It does not provide for velocity and assumes only elastic deformation, as well as perfection of surfaces and contours which, of course, is not attained practically. For this reason, it was desired to obtain a measure of the actual compression stresses involved in these tests by calculation from contact areas. The method of determining the contact areas was to plate one of each pair of test gears with 0.0008-inch of copper, place the pair in the machine under static load and to vibrate the system through a small amplitude for one hour. The contact area was thus well defined as a bright polished area on the copper plate. Because of imperfections on the surfaces, the contact areas were never of uniform width and frequently were discontinuous. More perfect surfaces of course produce more continuous contact areas of relatively narrower and uniform widths.

The teeth were then cut from the gears, photographed at magnification and the gross and net contact areas de-

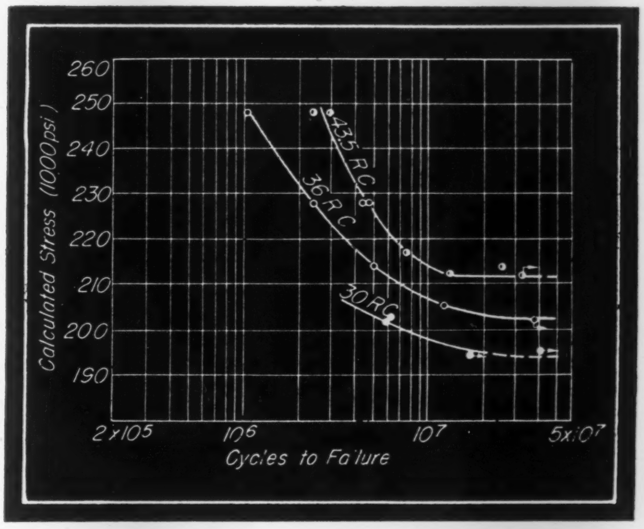
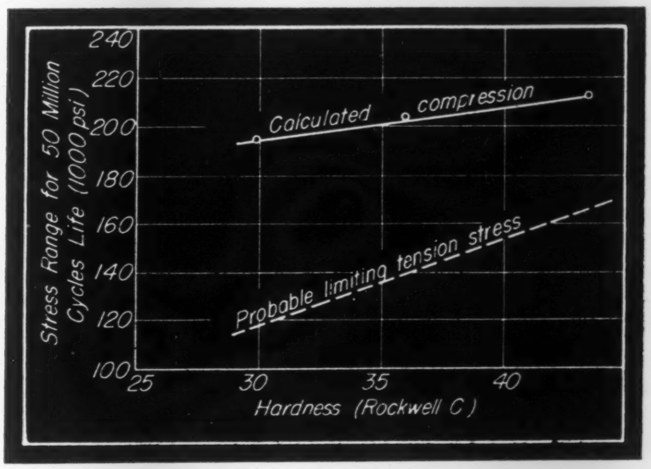


Fig. 7—Above—Stress-life curves for SAE 4645 gears tempered to three levels of hardness

Fig. 8—Below—Calculated compression and probable limiting tension stress ranges. Loading cycle ranges from zero to the maximum



[†] Developed by J. C. Straub, General Motors Research Laboratory.

termined by projecting the images on a cross-sectioned screen and counting occupied squares. The gross area was taken as the total area enveloping the numerous small contact areas and the discontinuities while the net area was the summation of the actual contact areas only. The average compression stress over the projected zone was then calculated by s_c = Load/Area, where load on tooth = ½ loading torque (in.-lb)/*PR*, and *PR* = pitch radius = 1.75 inches.

Results are plotted in *Fig.* 9. The average compression stress over the gross projected zones plot with considerable scatter as might be expected with imperfect surfaces. The mean of this scatter band, however, probably represents a close approximation of the true average stresses over the gross projected zones. When the areas actually bearing the loads, that is the net areas, are considered, the true average stresses approach those calculated by the formula more closely as the loads increase and imperfections are eliminated through plastic deformation. Within the range of loading torque used, the deviation from the formula stress is indicated to be about 15 per cent at 200 ft-lb, but decreases with increasing torque to about 3 per cent at 625 ft-lb. The suggested small but constant difference at higher loads may be the difference between the calculated maximum and the determined average stresses. Agreement between calculated and determined stresses over the full stress range should improve with improvement of surfaces and hardness.

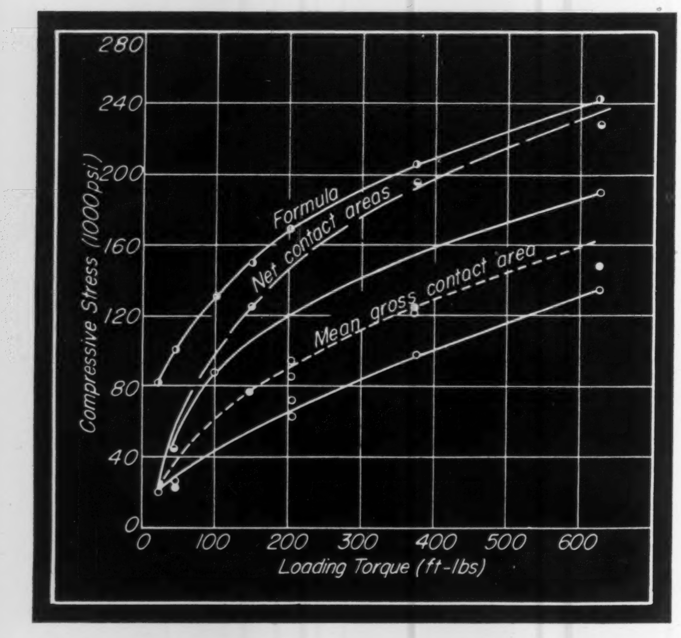
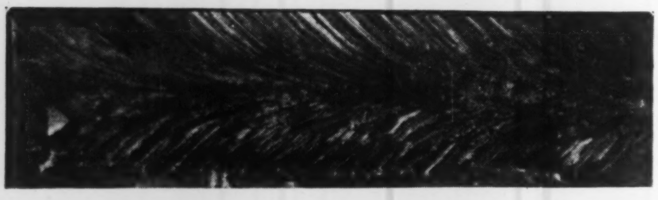


Fig. 9—Above—Comparison of surface stresses on gears calculated from formula and determined from contact areas

Fig. 10—Below—Badly scored gear tooth, at 3.785 magnification, illustrating direction of forces



Stresses calculated are compressive. According to Almen[‡] fatigue failures do not result from repeated compressive stresses but only from repeated tension stresses. The tension stresses which cause failure in these gear tests must be of a lower magnitude than the calculated compressive stresses. An idea of the values of the maximum safe repeated tension stresses for steels of these hardnesses is obtained from the lower curve of *Fig.* 8. This is the statistical relation under ideal conditions between hardness and the limiting safe stress for a stress range zero to maximum. These stresses are about 80 per cent of the tensile strengths and are about 60 per cent higher than the limiting safe stresses for complete reversal[§]. They probably represent the maximum safe values of the effective tension stresses involved in these gear tests.

Failure Stresses Complex

The stresses which initiate failure, however, must not be simple tension for they originate from two sources: (1) The tension component of the operating compressive stress, and (2) the tangential tensions in the teeth surfaces developed by the action of the teeth during operation. The second is believed to be of greater importance because of the conditions peculiar to this test. It will be recalled that the test gears rotate in different planes which include an angle of 15 degrees; because of this, each tooth slides endwise over its mate during the entire period of contact. Observations show that the first sign of tooth distress during test is a scoring action which develops a series of graceful "arrowhead" markings with their apexes on the pitch line pointing in the direction of the sliding, *Fig.* 10. This frictional force must develop a high degree of tangential tension stress in the tooth surface and it is evidently more severe in the driving tooth, for it always pits first and more extensively. This is probably associated with the fact that the surface metal in the driving gear is pulled away from the pitch line both during engagement and disengagement whereas, in the driven gear, the tendency is to move metal toward the pitch line.

Microscopic examination of incipiently failed gears indicated that, first, fatigue cracks form normal to the tooth surface as well as to the direction of the surface forces. Subsequently, a particle of metal is sheared out from this crack in the direction of the force to form a pit which in the driving gear is away from the pitch line and in the driven gear toward it. This is so characteristic that inspection usually reveals which of a pair of gears was driving and which was driven. Once started, the spread of pitting is rapid and destructive. At this stage the noise emitted increases continuously.

Use of this machine as a laboratory tool has been highly satisfactory and the work is being continued. As an ultimate aim it is hoped to be able to compare materials of gear construction and their treatments but it has been learned already that many factors must be closely controlled. The immediate problem is to specify a finish which can always be duplicated since this has been found to be of utmost importance.

[‡] *SAE Transactions*, Vol. 51, No. 7, Page 248.

[§] Moore and Kommers—"Fatigue of Metals", McGraw Hill Book Co., Page 192.

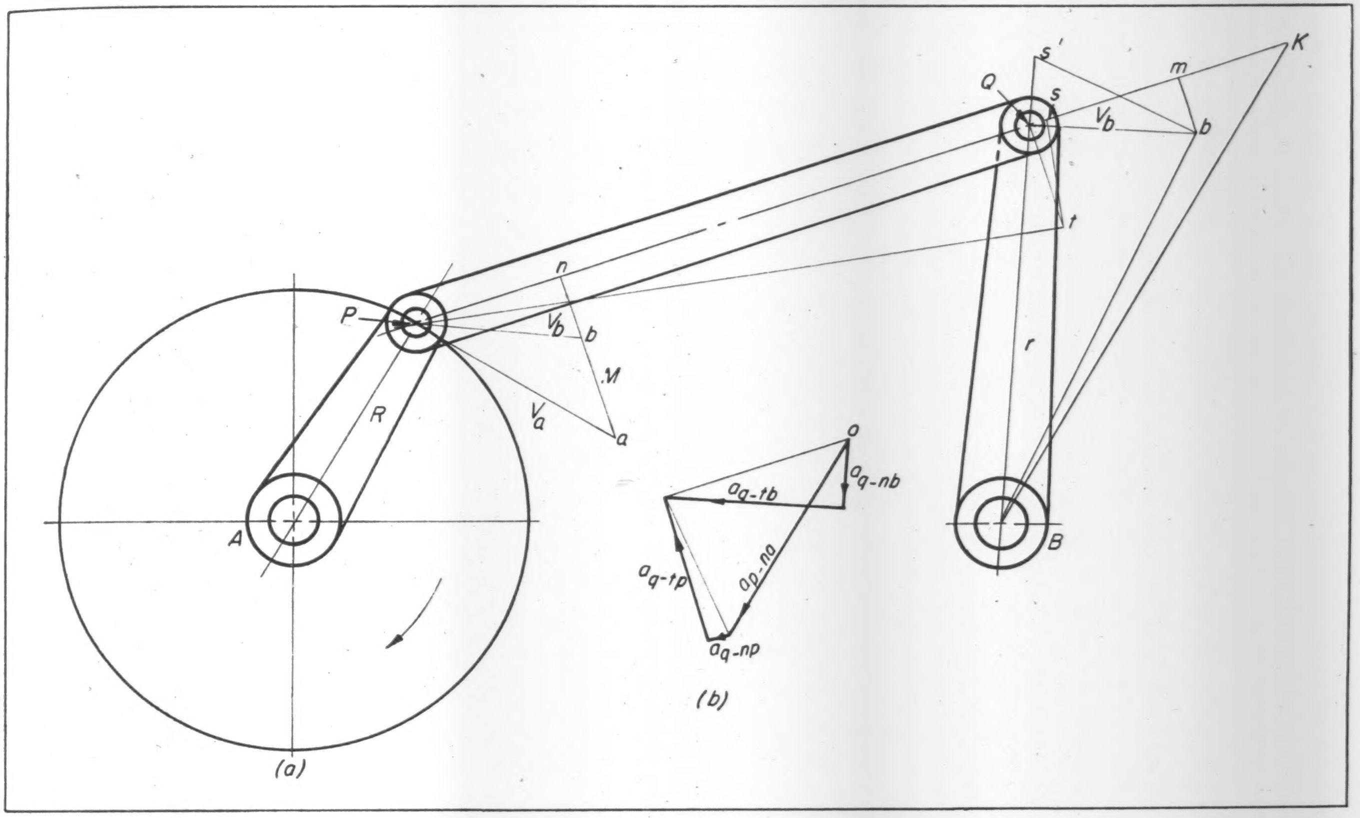


Fig. 1—Top—Graphical construction for finding velocities and accelerations in simple crank and rocker mechanism

How to Determine Accelerations

. . . for mechanisms in which key points move along paths that rotate

By Philip K. Slaymaker
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Lincoln, Nebr.

IN THE motion analysis of high-speed mechanisms there is frequently encountered the principle known by the somewhat forbidding terms "Coriolis' law" or the "compound supplementary acceleration", disregard of which may lead to serious error in calculating inertia stresses. It is therefore important that the designer should know how to recognize this condition and how to determine its magnitude.

When a point moves along a path while at the same time the path itself has an independent motion of translation only, the linear acceleration of the point is the vector sum of the acceleration of the point relative to the path and the acceleration of a coincident point in the path itself. However, if the path rotates, a third acceleration component must be considered. The principle involved in working out this third acceleration component is known as Coriolis' law.

With the idea of making this principle more readily understandable to the average engineer, this article demonstrates a graphical construction for finding the linear acceleration of the ram of a shaper. The ram is operated by means of the oscillating-arm quick-return motion.

In order to introduce the graphical construction, prior to the discussion of the more difficult quick-return motion, the simple construction for the crank and rocker mechanism shown in *Fig.* 1 will first be explained.

In *Fig.* 1*a*, *PA* is the driving crank rotating about the center *A*. The driven crank or rocker oscillates about center *B* and is connected to the driving crank *PA* by the link *PQ*. The length of *PA* is designated by *R* and the length of *QB* by *r*.

Point *P* has a tangential velocity V_a which may be represented by the vector *Pa* drawn perpendicular to the crank *PA* with length equal to *R*. The velocity V_a is resolved into components along and perpendicular to *PQ* by dropping the perpendicular *an* to *PQ*. *Pn* is the component of the velocity V_a along *PQ*, and *na* is the perpendicular component.

To find the tangential velocity of *Q*, an indefinite line is drawn perpendicular to *QB*. Since all points on the link have the same velocity component along the axis of the link, the component of *Q* along the link will be the same as that of *P*. Therefore the component *Qm* is laid off equal to *Pn* and a perpendicular erected to the link at *m*, intersecting the line drawn perpendicular to *QB* at *b*. *QB* is now marked V_b and represents the tangential velocity of *Q*. By transferring the triangle *Qbm* to the other end of the link and imposing *Qm* on *Pn*, a new triangle is formed whose sides are V_aV_bM. It should be noted that the length *M* (= *ba*) will always be either the sum or the difference of the perpendicular components of *P* and *Q* with reference to the link, and represents the tangential velocity of *Q* with respect to *P*. This new triangle is sometimes called the "velocity image". Also it may be noted that V_a is the vector sum of V_b and *M*.

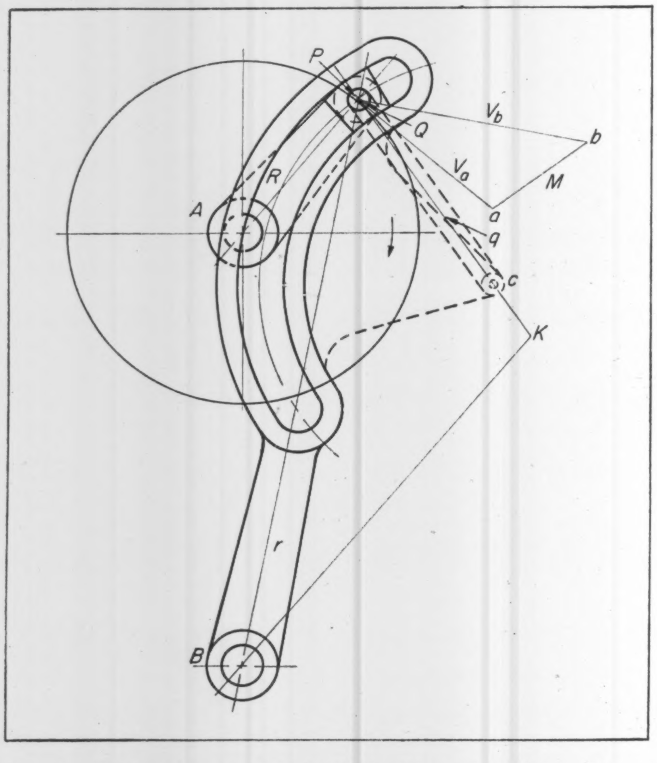


Fig. 2—Replacement of link PQ in Fig. 1 by sliding block and slot introduces difficulties in acceleration analysis

The point *K* in *Fig.* 1*a* should also be noted. If a line is drawn from the center *B* parallel to *R* it will intersect the line *PQ*, or *PQ* extended, at *K*. This point if considered as rotating with *r* will have the same velocity as *P* because the direction of its motion and its component of velocity along the link are the same as for *P*. The point *K* is therefore called the "point of identical velocity".

Accelerations are vector quantities and the acceleration of *P* will therefore equal the vector sum of the acceleration of *P* with respect to *Q* and the acceleration of *Q* with respect to *B*. However, each acceleration is made up of two components. When a point rotates about a center it may or may not have an acceleration along the tangent to the path of motion, but it will always have an acceleration toward the center of rotation, which is known as the normal acceleration. The total acceleration is the vector sum of the normal and tangential accelerations. If all these accelerations are plotted as vectors they will form a closed figure.

Referring again to *Fig.* 1, the acceleration polygon is plotted beginning with the driving crank, the rotative speed of which is known. The normal acceleration of any point rotating about a center with tangential velocity *V* and radius *R* is equal to V^2/R or ω^2R in which ω represents the angular velocity.

Scale for Accelerations

If the tangential velocity is represented by a line equal in length to the radius, then $V^2/R = R^2/R = R$. In this case, as in that of the velocity problem, a vector equal in length to the radius will represent to scale the normal acceleration of *P* as it rotates about *A*.

In *Fig.* 1*b* a line is drawn from a point *o* with a length equal to that of the radius *R* and in the direction of *P* toward *A*. This line is marked $a_{p\text{-}na}$, the arrow showing the direction of the acceleration. The symbol $a_{p\text{-}na}$ is read "acceleration of *P* normal to *A*". As stated before, if the accelerations are plotted in regular order they will form a closed polygon. Since the crank *R* is rotating at a constant speed the tangential acceleration of *P* with respect to *A* is zero. Hence $a_{p\text{-}na}$ is, in this case, the total acceleration of *P*.

Next step is to plot from the terminus of the vector just drawn the total acceleration of *Q* with respect to *P*. This total acceleration is made up of two components, the normal and the tangential accelerations. The normal acceleration of *Q* with respect to *P* will be M^2/QP. The length of this vector is found graphically as follows: On *Fig.* 1*a*, *Qt* is drawn perpendicular to *QP* with length equal to *M*, *P* connected with *t*, and *ts* drawn perpendicular to *Pt*, intersecting *PQ* extended at *s*. *Qs* will be the length of the required vector. This can be readily seen for, by similar triangles, $Qs/Qt = Qt/QP$ or $Qs = (Qt)^2/QP$, but since $M = Qt$, $Qs = M^2/QP = a_{q\text{-}np}$. In *Fig.* 1*b* this value is plotted from the terminus of $a_{p\text{-}na}$ in the direction of *Q* toward *P*. The tangential acceleration of *Q* with respect to *P* will be perpendicular to the normal component but as there are no other data concerning this tangential component, an indefinite line is drawn at the terminus of $a_{q\text{-}np}$ and perpendicular to it.

Since no further progress can be made in this direction, but it is known that the polygon must close at the point *o*, the next step will be to plot the acceleration of *Q* with respect to *B* from the point *o*. The normal acceleration is V_b^2/r and the length of this vector is found in the same manner as for Q with respect to P. The line Bb is drawn connecting the center B with the end of the velocity vector V_b. Now a perpendicular to Bb is erected at b, intersecting BQ extended at s'. Then Qs' is the length of the required acceleration vector. In *Fig.* 1*b* this vector is drawn from o in the direction of QB and marked $a_{q\text{-}nb}$, the arrow again showing the direction of the acceleration.

To find the tangential acceleration of Q with respect to B, a line is drawn at the terminus of $a_{q\text{-}nb}$ and perpendicular to it. This line will intersect the indefinite line drawn perpendicular to $a_{q\text{-}np}$ and thus close the polygon, fixing the lengths of the vectors $a_{q\text{-}tp}$ and $a_{q\text{-}tb}$. The latter value is the one required, viz., the acceleration of Q in its path as it oscillates about B. The direction of the arrow will show the direction of the acceleration.

To find the acceleration polygon in the case of the crank and slotted rocker shown in *Fig.* 2, the same procedure as the foregoing is followed. In this figure there is again the driving crank PA rotating about a center A, but in this case the crankpin P is compelled to move in a slot with a radius Pc. This slot is a part of a piece rotating about a center B, but the point Q of the slotted piece is always coincident with the point P of the driving crank. In other words, the link PQ of *Fig.* 1*a* has become zero in length in *Fig.* 2. However, it will still be necessary in this new mechanism to find the acceleration of Q with respect to P in order to complete the acceleration polygon. It will be easier to find the acceleration of P with respect to Q rather than that of Q to P. The value of the acceleration will be the same but its direction will be reversed.

Equivalent Linkage Aids Concept

Using the same notation in *Fig.* 2 as in *Fig.* 1, the velocity triangle is constructed. Vector V_a represents the velocity of P as it rotates about A with radius R. V_b represents the velocity of Q as it rotates about B with radius r. M, the velocity of P with reference to the slot, is drawn perpendicular to the radius of the slot for the position shown. The radius of the slot has its center at c. Since the point P moves in the slot its motion will be the same as if it were attached to a link rotating about the point c, therefore the link Pc can be substituted for the slot and its radius designated q.

The normal acceleration of P with respect to Q must equal the normal acceleration of P with respect to c minus the normal acceleration of Q with respect to c. Putting this in the form of an equation,

$$a_{p-nq}=a_{p-nc}-a_{q-nc} \quad \text{(1)}$$

but

$$a_{p-nc}=\omega_q^2q$$

and

$$a_{q-nc}=\omega_r^2q$$

where the angular velocity of $r=\omega_r$, and the angular velocity of $q=\omega_q$. Substituting these values in Equation 1,

$$a_{p-nq}=\omega_q^2q-\omega_r^2q \quad \text{(2)}$$

But the absolute value of ω_q is

$$\omega_q=\omega_r+\omega_{q-r} \quad \text{(3)}$$

in which $\omega_{q\text{-}r}$ is the angular velocity of q with respect to r. In applying Equation 3 particular attention must be paid to the sign of each term, that is, whether both velocities are in the same sense or in opposite sense. Also, $\omega_{q\text{-}r}=M/q$. Substituting this value in Equation 3,

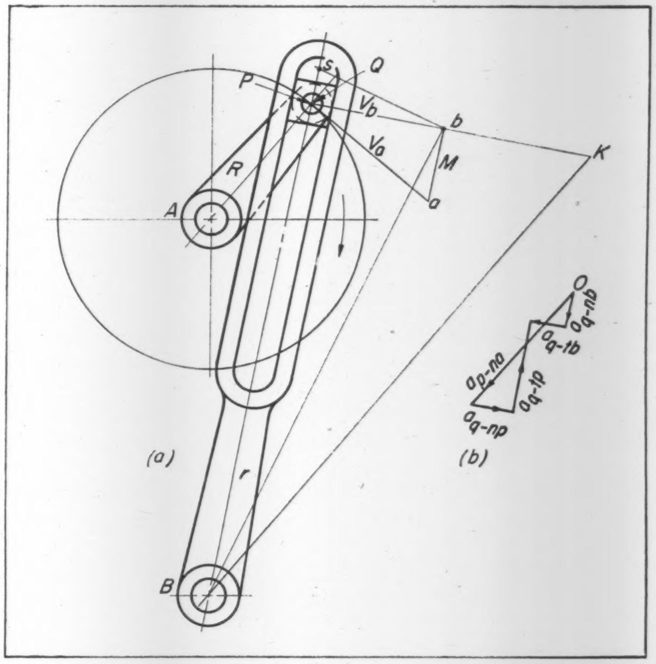


Fig. 3—In practical shaper quick-return mechanism the curved slot of Fig. 2 becomes straight. Analysis is similar

$$\omega_q=\omega_r+\frac{M}{q} \quad \text{(4)}$$

Substituting this value of ω_q in Equation 2, it becomes

$$\overline{a_{p-nq}}=\left(\omega_r+\frac{M}{q}\right)^2q-\omega_r^2q$$

$$=\frac{M^2}{a}+2\omega_rM \quad \text{(5)}$$

The principle involved in the "compound supplementary acceleration" $(2\omega_rM)$ as finally expressed in Equation 5 is known as "Coriolis' Law". The normal acceleration of P with respect to Q always acts as *from* the point of identical velocity K, which is shown in *Figs.* 1*a* and 2. The acceleration of Q with respect to P has the same value but acts in the opposite direction.

That the acceleration of P with respect to Q acts from the direction of point K can readily be seen by referring to *Fig.* 2. In this case the points P (on the crank) and Q (on the slotted piece) coincide, but each has a definite normal acceleration toward the center of rotation c. The difference between these values is the required acceleration of P normal to Q. It will be noted from *Fig.* 2 that the direction of motion of the sliding block will cause q to rotate in opposite sense to r, that is, ω_{q-r} is opposite in sign to ω_r. From Equation 3, ω_q therefore is numerically less than ω_r and the normal acceleration of P toward c is less than that of Q toward c (Equations 1 and 2). This gives the relative acceleration $a_{p\text{-}nq}$ as coming *from* K. The acceleration $a_{q\text{-}np}$ of course will be *toward* K as in the case of *Fig.* 1.

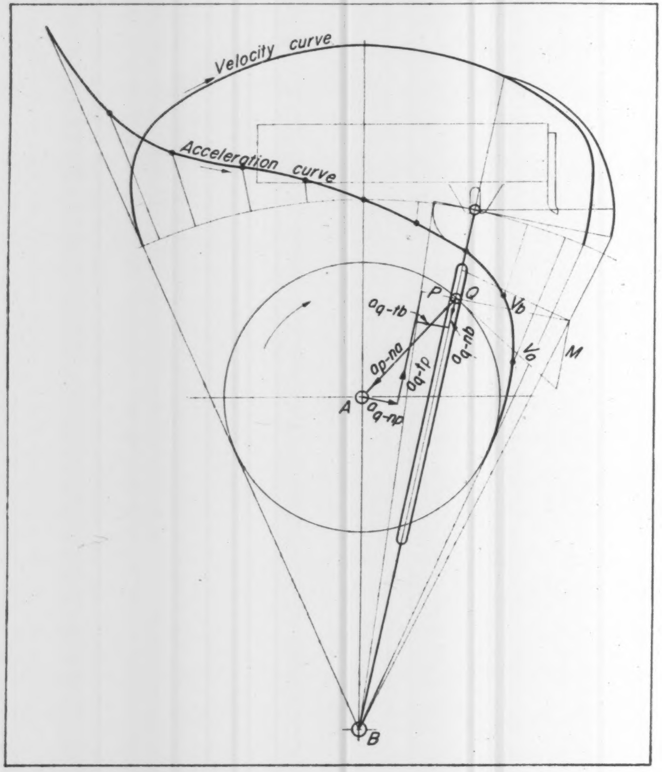


Fig. 4—Plot of the velocities and accelerations for an oscillating-arm quick-return mechanism for a complete cycle. Values represent tangential motion of the arm

On the other hand, if the mechanism rotates to a position that will reverse the direction of travel of P, this will make ω_q greater than ω_r and will reverse the direction of the relative acceleration. In such a position, however, the point K will shift to the other side of r, hence the rule will still apply.

In most cases the slot is straight, in which event the radius q becomes infinity and Equation 5 becomes

$$a_{p-nq}=2\omega_r M=2\frac{V_b}{r}M \qquad (6)$$

To apply the foregoing case of the pin and slotted rocker as usually found in the quick-return motion of a shaper the procedure is as follows: In *Fig.* 3*a* is shown the same arrangement as *Fig.* 2 except that the slot is straight. The notation is the same as for *Fig.* 2. From any point o in *Fig.* 3*b* as an origin, draw a line equal to R and in the direction of P to A. This line is marked $a_{p\text{-}na}$, the direction of the arrow indicating the direction of the acceleration. The next step is to plot from the termination of this line the normal acceleration of Q with respect to P as was done in *Fig.* 1*b*. The direction, however, in this case must be toward the point of identical velocity, K. The length of this line is found by the expression $2(V_b/r)M$ from Equation 6. This product is obtained by scaling the lines which represent V_b, r and M. This line is marked $a_{q\text{-}np}$, the arrow showing the direction of the acceleration. The tangential acceleration of Q with respect to P is unknown except that its direction will be perpendicular to $a_{q\text{-}np}$, therefore at the terminus of $a_{q\text{-}np}$ an indefinite line is drawn having this direction.

Normal Acceleration Found Graphically

Since the acceleration polygon must close at o the next step is to plot from o the normal acceleration of Q with respect to B, which is equal to $V_b{}^2/r$. This value may be obtained by scaling the lines in *Fig.* 3*a* or by graphical construction as follows: Draw a line from B to the end of the velocity vector V_b at b. Erect a perpendicular to this line at b to intersect BQ extended at s. The line Qs will then be the value of the required acceleration.

This length is plotted from o in the direction of QB and marked $a_{q\text{-}nb}$, the direction of the acceleration being shown by the arrow. At the termination of this line an indefinite line is drawn perpendicular to it to meet the similar line drawn perpendicular to $a_{q\text{-}np}$. This line is the required tangential acceleration of Q as it rotates with radius r about B, and its direction is such as to close the polygon. It is marked $a_{q\text{-}tb}$. The closing of the polygon also determines the value of the acceleration of Q relative to P along the slot, $a_{q\text{-}tp}$.

To find the actual value of $a_{q\text{-}tb}$, it will be remembered that $a_{p\text{-}na}$ was drawn equal to the radius R. Knowing the angular velocity of R and the length of the radius, $V_a{}^2/R$ can readily be calculated. V_a in the foregoing case is the velocity in feet per second, and R is the radius in feet. The length R will represent to scale the value just calculated. All other accelerations will be in proportion to the lengths of their vectors.

Accelerations Proportional to Radius

The connection between the rocker and the ram of the shaper is usually at some point along the rocker and beyond the slot. The acceleration of this point is in direct proportion to the length of the radius from the center B. However, if the ram is connected to this point by means of a link then the procedure will be similar to that of the crank and rocker (the rocker now being the driver), the accelerations of which have just been determined.

In *Fig.* 4 is shown the construction for velocity and acceleration curves for the cutting stroke of a shaper where the ram is connected directly to the rocker. The curves in this figure show the velocity and acceleration along the arc, but the values can be readily resolved along the direction of motion of the ram.

MACHINE Editorial DESIGN

Trade or Profession?

BECAUSE of his unique position in industry, the designer is not closely identified either with management or with labor. Although he is an employee working under the direction of management he must, within his field of authority, exercise the independent thought and judgment characteristic of a profession.

Designers, therefore, should consider carefully the possible implications in allowing labor unions to organize them into collective bargaining groups which include nonprofessional workers. In this movement there is danger of lessening the prestige which designers have earned over a period of many years.

That engineers, under the law, need not be compelled to join a union is now well established; any group has the right to set up its own independent collective bargaining unit through an election. This unit need not be affiliated with a national or local union and is under no compulsion to engage in the activities typical of a union. Under such a set-up designers have an opportunity to preserve their status as professional workers.

To maintain and enhance this status, however, they need some kind of an organization on the professional level that extends beyond the group in their own plant. Some designers may choose registration, which accords legal recognition of their standing as professional engineers. Others may find that active participation in national and local engineering society meetings contributes much to their personal and professional development, with consequent enhancement of their status.

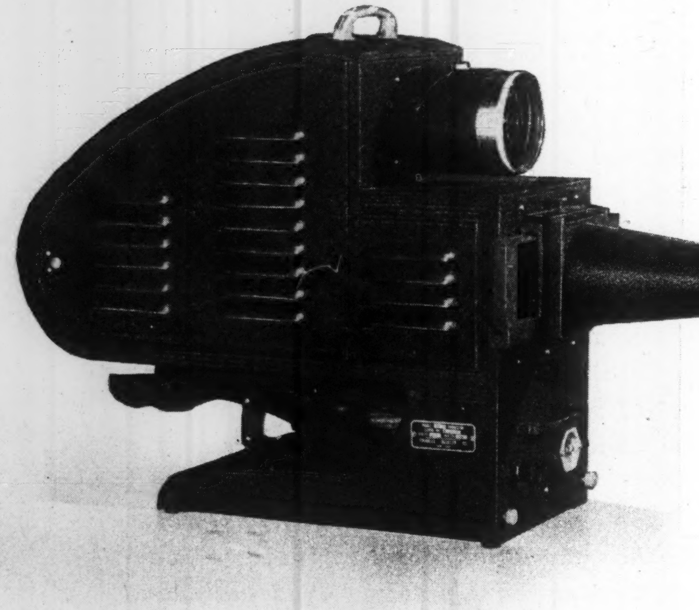
The recent formation of a Machine Design group within the American Society of Mechanical Engineers—previously reported in these pages—offers an opportunity for designers to associate themselves with an organization devoted specifically to their interests. This group held its first technical session in Detroit last month, and the size and caliber of the attendance gave ample evidence of the need for such an organization.

It is earnestly hoped that designers will give careful thought to the future of their chosen calling, resist the temptation to align themselves with selfish pressure groups, and jealously guard their present high professional standing.

Outstanding Designs

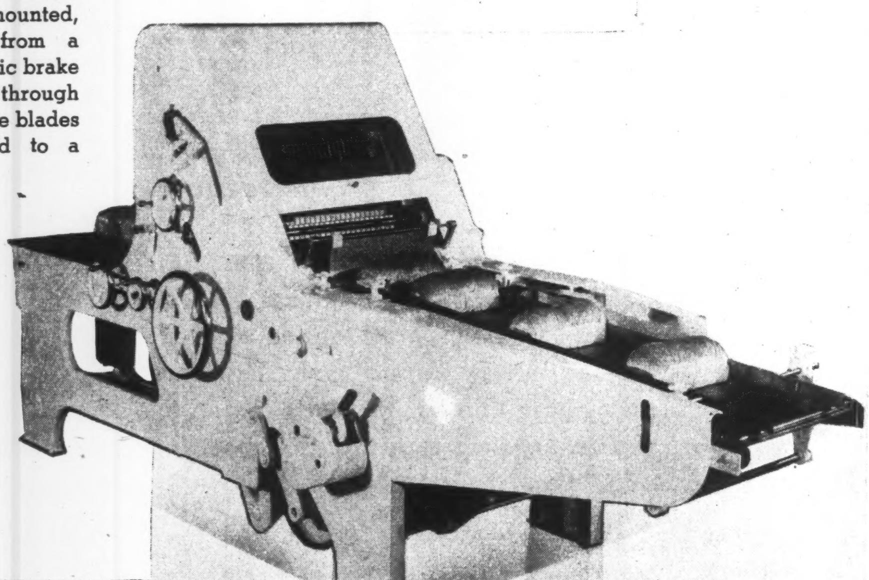
Projector

Capable of projecting opaque copy such as photographs, blue prints, book pages, etc., as well as standard lantern slides, this machine is fabricated almost entirely of sheet metal. Illuminant used is a 500-watt projection lamp which, with the aid of a specially formed parabolic reflector, directs a uniform beam of light upon the copy. Large aperture optical systems and a first-surfaced evaporated aluminum mirror are utilized to provide brilliant, sharp screen images. All controls are accessible from the exterior and two large doors provide access to the interior for servicing. Cooling is effected by a motor-driven centrifugal fan. Exterior finish is black wrinkle. Lens barrel and other trim parts are in polished chrome. Manufacturer: Charles Beseler Co., New York.



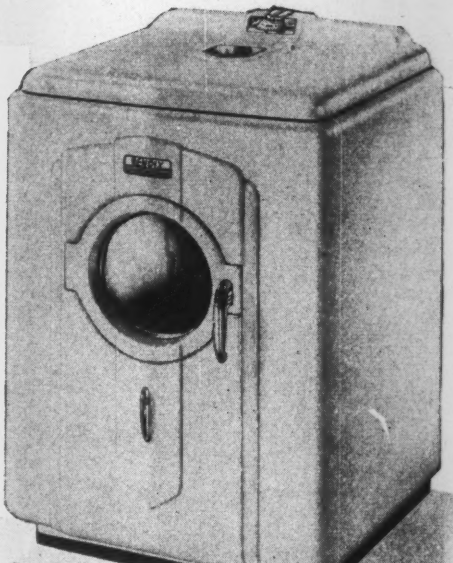
Bread Slicing Machine

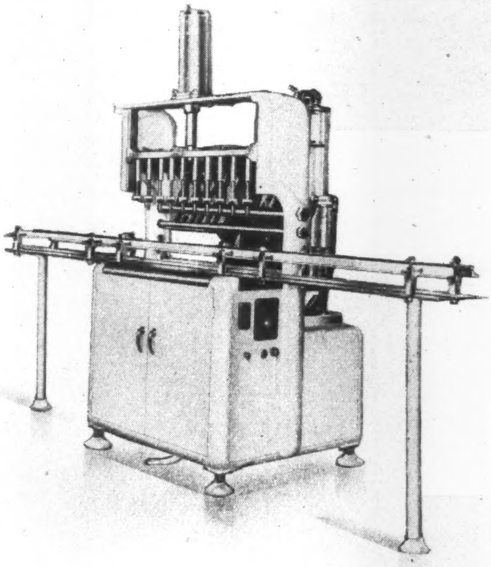
Bread slicing is effected in this machine by 42 chromium-steel endless blades formed in a figure eight, crossed at the slicing zone and operated over upper and lower drums which drive the blades. Blade guides are hard chrome-plated tool steel employed as inserts in zinc-alloy die castings. Steel drums, ball-bearing mounted, are double-V-belt driven from a motor incorporating a magnetic brake which is shut off automatically through a pair of bus bars spanning the blades and connected by solenoid to a throw-out switch. To prevent unnecessary blade breakage, an automatic blade-tensioning device is used in upper and lower drums together with an overload ratchet. Manufacturer: Battle Creek Bread Wrapping Machine Co., Battle Creek, Mich.



Home Laundry

Entire operating cycle of this machine, including washing, rinsing and damp drying, is controlled by a synchronous-motor-driven timer which opens and closes contacts by means of a set of cams. These contacts control the drive motor and also operate solenoid valves which, in proper sequence, actuate automatic mixing valves, drain valve, and a two-speed transmission. The horizontal-axis type perforated cylinder and surrounding tub are steel, coated with porcelain enamel. Top of cabinet is finished in porcelain; the remainder in high-temperature baked synthetic enamel. Manufacturer: Bendix Home Appliances Inc., South Bend, Ind.



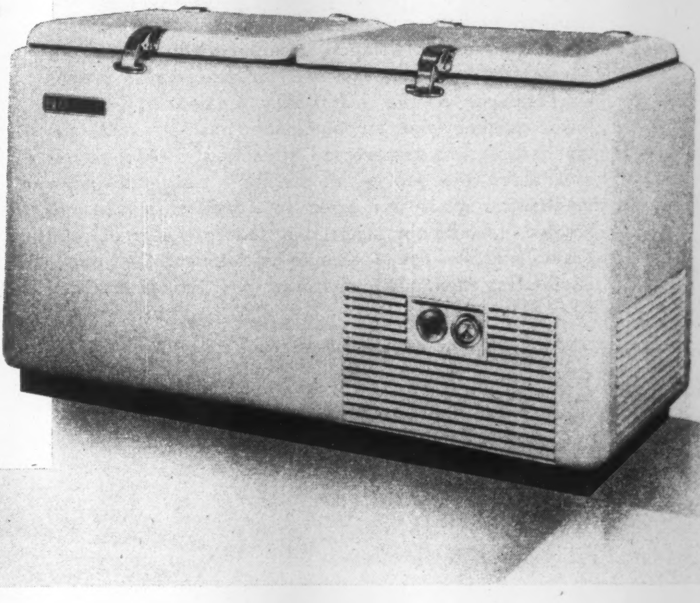


Vacuum Bottle Filler

Suitable for rapid filling of bottles and jugs within the height range of 3 to 13 inches, this machine is controlled by a balanced foot valve to permit the operator free use of his hands. Dripless valve-type spouts are employed and all liquid-contacting parts are either bronze plated or stainless steel. Automatically lubricated vacuum pumps, along with selector valves and motor, are fully enclosed in the cabinet. Pilot lights indicate operative position of the vacuum circuits. Manufacturer: Ertel Engineering Corp., Kingston, N. Y

Freezing Unit

With one frozen-storage and one fast-freezing compartment, this freezer has an all-steel, welded cabinet finished in baked white enamel. Covers are sealed with double extruded tubular rubber gasketing. Thick fiber-glass insulation employed is hermetically sealed to prevent moisture and air from changing insulation qualities. Refrigerant coils are copper, bonded to sides, ends and bottoms of freezing compartments to insure efficient heat transfer. Forced-flow, large capacity, air-cooled condenser is employed. Manufacturer: Ben-Hur Mfg. Co., Milwaukee.



RAYTHEON
precipitator

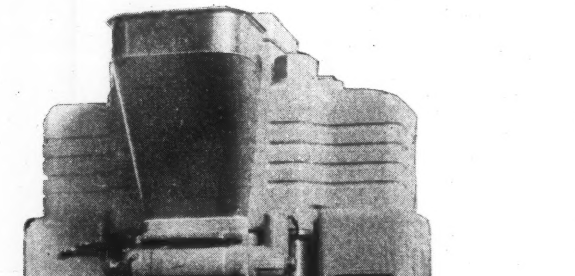
Precipitator

This machine recirculates air and removes 90 per cent or more of all contaminating particles of dust, smoke, oil mist, lint or pollen from the air. Primary components of the machine are: Blower, power pack, ionizer and electronic collector cell. The power pack rectifies a-c to d-c and steps up the 115-volt line supply to 12,000 volts for the ionizer and 6000 volts for the collector. As the dust-laden air passes through the ionizer, its electrostatic field imparts a positive charge to all dust particles. Air then passes through the negatively charged collector plates which collect the dust particles, permitting only clean air to pass out of the machine. Collector plates are aluminum and are suspended vertically from tie rods. Manufacturer: Raytheon Mfr. Co., Waltham, Mass.

Molding Press

All control mechanisms of this machine—electric, air and hydraulic—are built into the two side frames and removable full length panels make every control readily accessible. Mechanism for measuring powder charges is multiple-station, pneumatically controlled. Thermostatic controls are used for maintaining proper temperatures of the electrically heated upper and lower platens and an automatic push-off combines mechanical and compressed air action to eject pieces and clean the mold. A sensitive weight-checking mechanism stops the press in event parts are not ejected, insufficient material is fed to the mold, etc. Frame members are Meehanite castings and the hopper is stainless steel. Manufacturer: F. J. Stokes Machine Co., Philadelphia.

(New machines listed on Page 190)



Square Root of Sum of Squares

By Carl P. Nachod
Vice President
Nachod & United States Signal Co. Inc.
Louisville, Ky.

CALCULATION of the square root of the sum or difference of two squares—a frequent necessity in engineering work—is a somewhat tedious procedure due to the combination of squaring, adding or subtracting, and extracting the root. To speed up the calculation the nomograph which appears on the next page of this Data Sheet will be found useful and sufficiently accurate for most purposes.

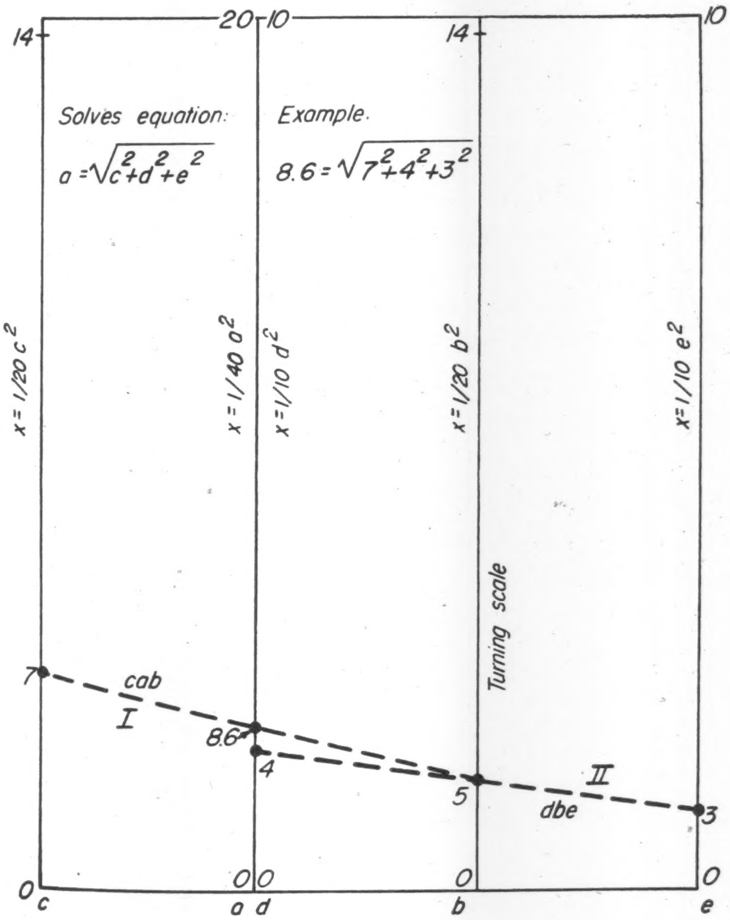
Designed to solve the equation

$$a=\sqrt{b^2+c^2}$$

the nomograph may also be used for differences. Since $a^2=b^2+c^2$, by transposition $c^2=a^2-b^2$ or

$$c=\sqrt{a^2-b^2}$$

Fig. 1—Basic dimensions for laying out a nomograph for the square root of the sum of squares. The four scales are equidistant



To extend the range of the nomograph it is only necessary to multiply or divided each of the three quantities by the same multiple of 10, or any other number. Thus the secant on the chart shows, for example,

$$8.6=\sqrt{5^2+7^2}$$

or

$$.86=\sqrt{0.5^2+0.7^2}$$

or

$$86=\sqrt{50^2+70^2}$$

or

$$4.3=\sqrt{2.5^2+3.5^2} \qquad \text{etc.}$$

When three or more squares are involved, the result of two squares may be transferred to one of the outer scales, then a new square taken. For example, if it is required to find the value of

$$\sqrt{5^2+7^2\;+9^2}$$

the 8.6 on *Fig.* 2 may be transferred to the *c*-scale and the value 9 located on the *b*-scale. A secant joining these two values would give the answer 12.45.

If three squares are to be considered frequently, it is quicker and more accurate to add a set of graduations to scale *a*, as well as a new scale. A sketch of the resulting nomograph is shown in *Fig.* 1, the new scale being labeled *e* and the additional graduation *d*. Actually the turning scale, *b*, need not be graduated. The two secants solve, respectively, the equation

$$a=\sqrt{b^2+c^2} \qquad \text{(by secant } cab\text{)}$$

and

$$b=\sqrt{d^2+e^2} \qquad \text{(by secant } dbe\text{)}$$

from which *b* may be eliminated, leaving

$$a=\sqrt{c^2+d^2\;+e^2}$$

To use the 4-line chart, *Fig.* 1, the secant *dbe* connecting *d* (=4) and *e* (=3) locates the point on the turning scale from which secant *cab* is drawn to *c* (=7). The intersection with scale *a* (=8.6) is the required answer.

Reproduction of *Figs.* 1 and 2 may be easily accomplished using a table of squares and an engineer's scale divided into 10ths, 20ths and 40ths. The equations of the several scales are shown in *Fig.* 1, in which *x* is the distance in inches from the base. All scales are equidistant and graduations begin at the base. The advantage of *Fig.* 1 is that it does not require the reading to be transferred to another scale, which takes time and is a possible source of error.

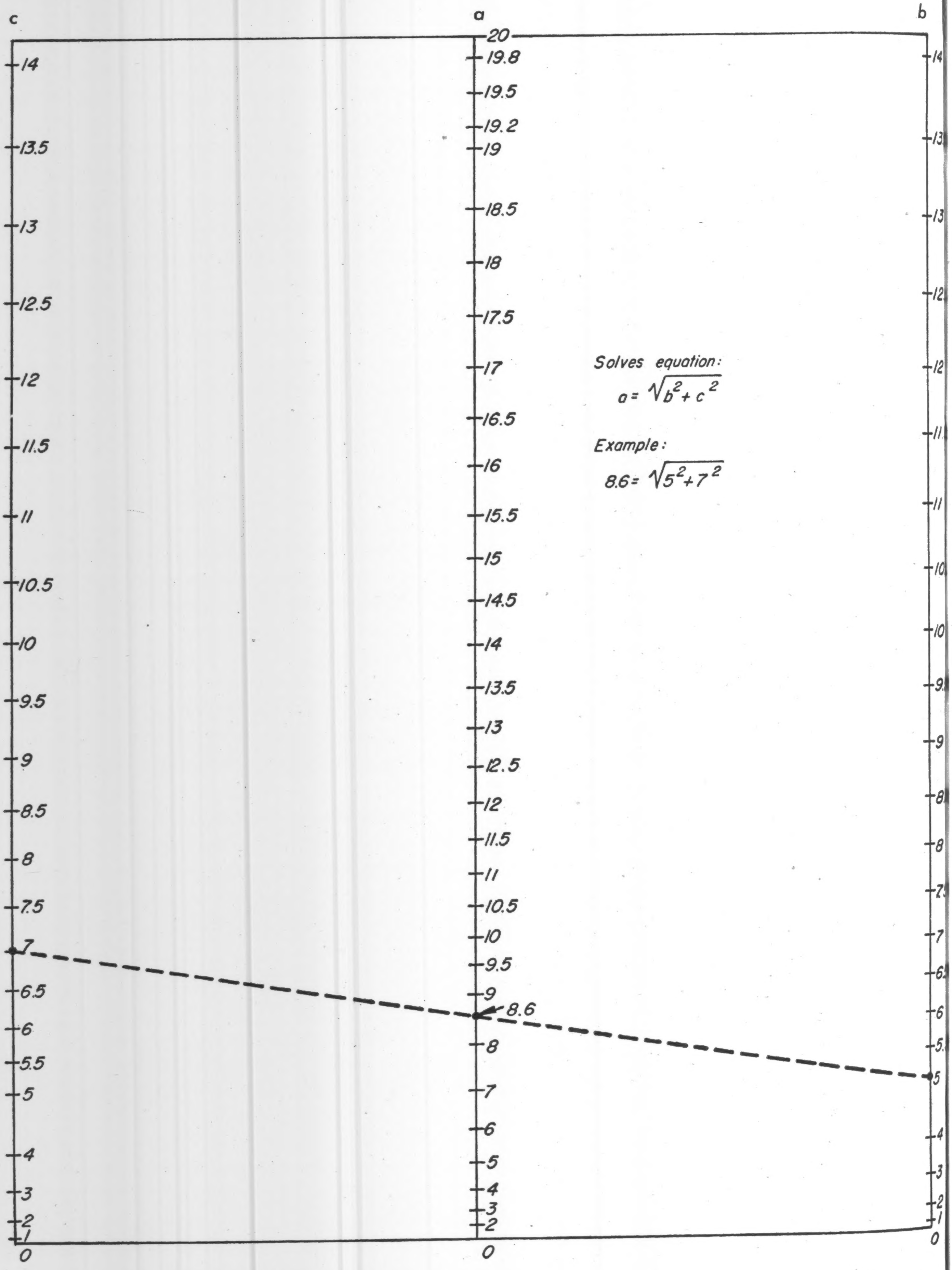


Fig. 2—Nomograph for determining the square root of the sum of two squares. Answer is found on the middle scale

Noteworthy Patents

FUEL INJECTION PUMP IS ELIMINATED by a new hydraulic injector for compression-ignition engines covered in patent 2,389,492 assigned to Timken Roller Bearing Co. by H. C. Edwards. Actuated solely by the pressure of the air compressed in the engine cylinder, the device utilizes the pressure of by-passed fuel to determine the exact quantity of fuel delivered to the cylinder.

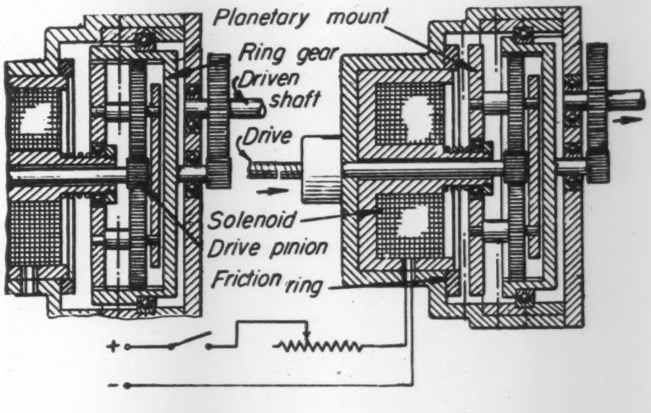
LOST MOTION IN JACK SCREWS or lead-screw mechanisms is effectively obviated with a nut device assigned to Lockheed Aircraft Corp. by George A. Caroll under patent 2,385,194. Particularly adaptable to machine tool feed mechanisms, cowl flap jacks, tab operating devices, etc., the arrangement operates automatically under all conditions to prevent backlash due to wear whether the nut is stationary or traveling. Adjusted simply at assembly, the compensating means remains effective until the screw threads have worn to approximately 25 per cent of their original size, whereupon the unit becomes loose to serve as a tell-tale.

POSITIVE CLUTCH MECHANISM for effecting an automatic shift between two of the gear ratios of a transmission is covered by patent 2,384,439 recently assigned to the Borg-Warner Corp. by H. E. Carnagua and K. K. Stough. Shift from a low ratio to a high ratio and vice versa is made automatically in response to a decelerating or accelerating torque load, respectively, by means of a positive jaw clutch mounted on a helically splined shaft. The clutch member, when automatically engaging in either ratio, energizes both a frictional synchronizing mechanism and a positive locking means.

EQUALIZED TRAVEL of two or more spaced hydraulic rams is achieved by means of a novel valve arrangement covered by patent 2,377,007. Assigned to the American Steel Foundries by Relza J. Hess, this simple unit embodies a valve and control lever so arranged that any tendency for one ram to move ahead of another is immediately and directly offset by the action on the valve of the lever which is attached to the moving platen. Change in proportion of fluid directed to the rams is accomplished without effecting any change in capacity of the valve. Owing to increased effective length of the actuating lever as the rams descend, accuracy and sensitivity increases reaching a maximum at final position.

TWO-SPEED DRIVE and clutch assembly which utilizes a novel ball clutch speed-changing mechanism is covered by patent 2,377,575 assigned to Nineteen Hundred Corp. by Luther Ringer. Adaptable primarily to light domestic machines such as ironers, the device is simple to operate, reliable and economical to manufacture.

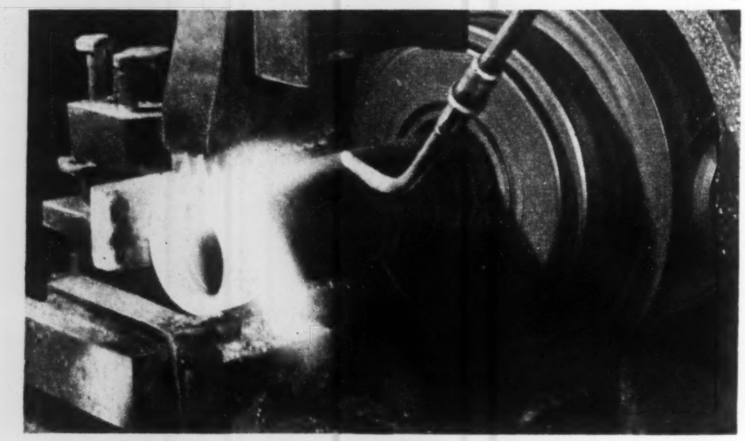
SELECTIVE SPEED CONTROL by means of infinitely variable magnetic restraint of planetary action is covered by patent 2,385,630 assigned to Lear Inc. by William P. Lear. Control of a magnetically actuated friction plate by adjustment of a rheostat creates in effect a restrained or controlled rotation of planetary gearing, thereby varying the ratio between driving and driven speeds over a wide range.



A listing of expiring patents is now available to those interested. This service is being set up as a weekly feature by the Scientific Development Corp., 614 West 49th St., New York 19, from whom details can be obtained by writing direct.

Applications

of Engineering Parts, Materials and Processes

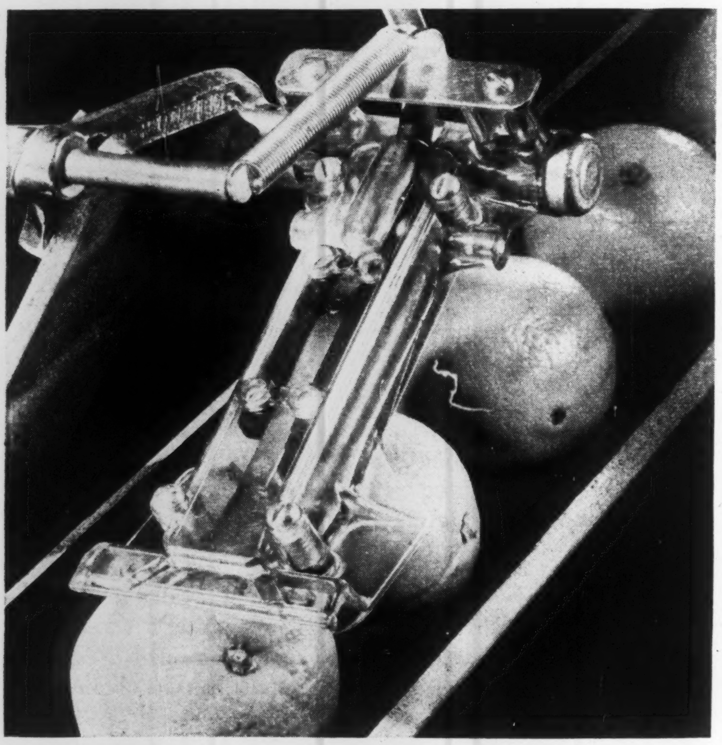


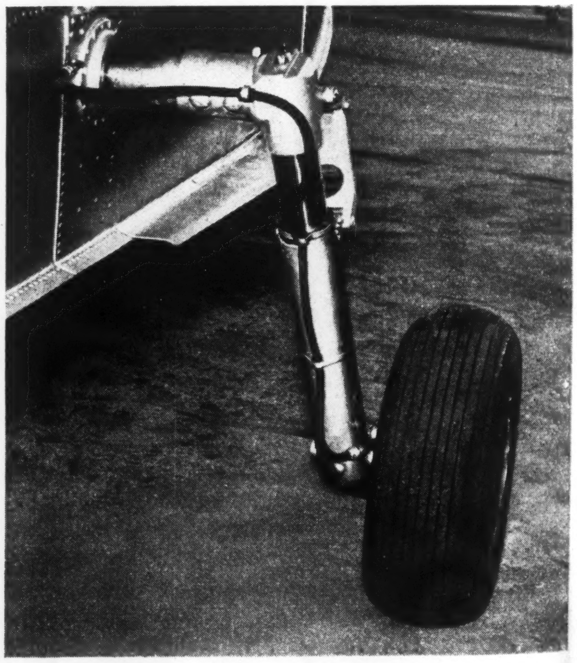
Flame Spinning

TUBE CLOSURES as well as diameter reductions to any particular configuration are made possible by the Linde Process of flame spinning left. Limited only by the ratio of tube diameter to wall thickness, which should not exceed 50 to 1, the process provides a rapid method of forming thick-walled cylinders. Essentially, the technique consists of heating the tube with oxy-acetylene flame and forming into the desired shape while rotating. Inasmuch as heating and forming must be performed simultaneously, the flame heads, the lubricant feeder and the forming tool are mounted upon a common carriage. Spinning tools used are either cams or formed rollers.

Plastic Design Reduces Inertia

USED IN the citrus fruit industry, counter paddles, shown below, are molded of Tenite (cellulose acetate butyrate) plastics material. The paddle, which touches the fruit as it travels down the belt, must possess light weight and toughness to prevent bruising, and also to prevent bouncing after being struck by the fast-traveling fruit. Weighing only 2½-oz., the counter contains electrical apparatus to record the number of units passing by.



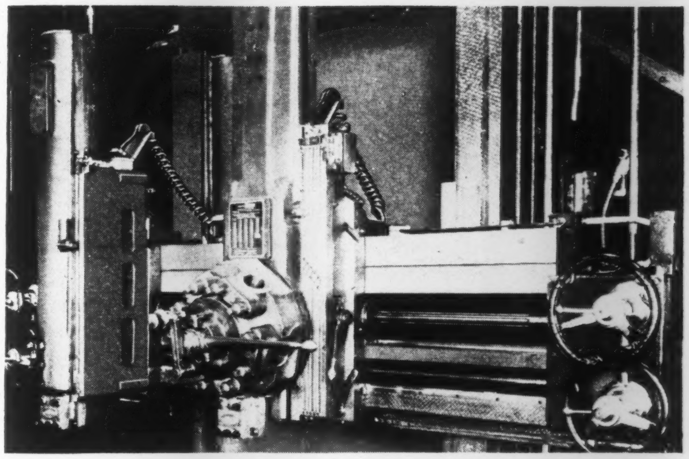


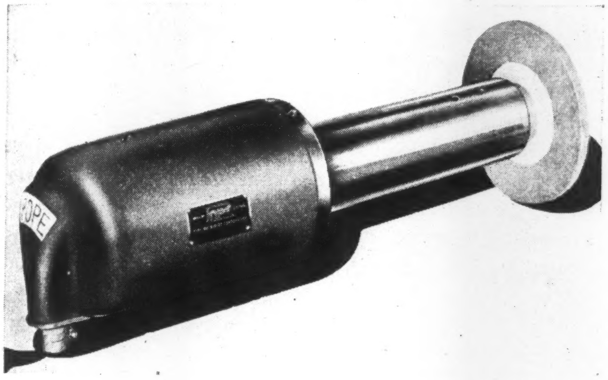
Unique Hydraulic Cylinder

SLIDING CONTACT between the piston and the cylinder wall is eliminated in the landing gear cylinders, above, by the use of stationary bearings located in the lower half of the cylinder. Piston contact being thus avoided, the cylinder interior need only be broached—honing is not necessary. The Electrol landing gear is a straight-leg, full cantilever design which is used on fixed or retractable undercarriages.

New Wire Makes Cleaner Design

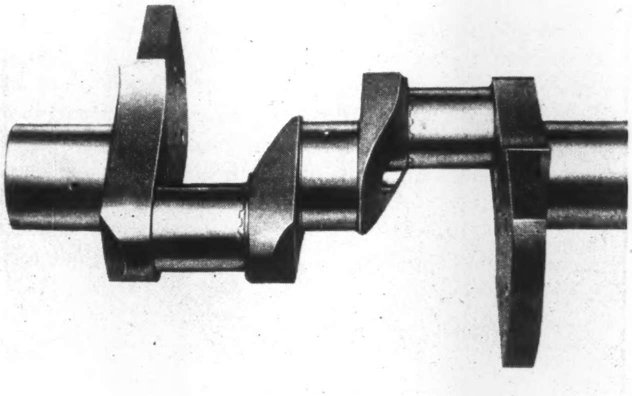
ELIMINATING the danger of loose, dragging wires, control circuits are connected in the vertical turret lathe shown at right by means of Kellogg Kords. Both turret head and ram have hydraulically operated stops actuated by two permanently positioned solenoids controlled by micro-switches mounted on the carriage at each head. Solenoids and switches are connected by Koiled Kords which stretch and retract to suit the horizontal or vertical traverse of either head. Use of neoprene-covered spring wire reduces the danger of operational interference which might be caused by tangling or looping of conventional type cords.





Sealed Unit Has Increased Life

MAKING the motor an integral part of the grinder spindle and increasing the service life, the Pope unit, left, eliminates many problems of dust and lubrication. The construction simplifies adjustment in the complete machine.

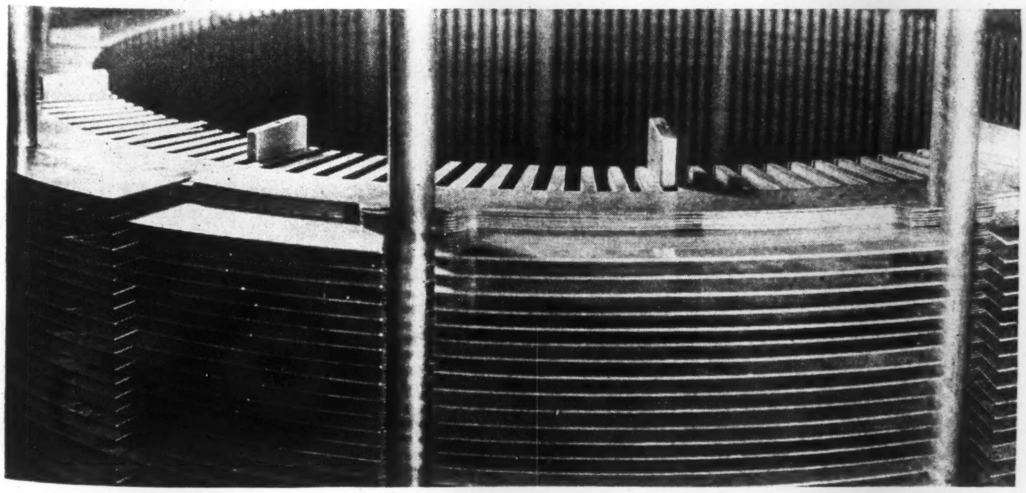


Coupling Simplifies Design

SIMPLIFYING design, expediting manufacture, the Gleason couplings in the two-throw crankshaft, left, used in a radial aircraft engine, make possible the use of solid connecting-rod bearings in place of the split type otherwise required. The couplings have radially-ground teeth of precise construction, the crankshaft assembly was made from completely interchangeable parts. In addition to transmitting torque, the coupling serves to align the parts, and other means of centering are unnecessary.

Cooled by Copper Fins

SUBSTITUTING copper laminations for air passages in motors, Westinghouse engineers have simplified the problem of designing totally enclosed fan cooled motors. Utilizing the fact that copper is a ten times better conducter than iron, stator punchings are made of thin copper sheets exactly like the iron punchings, but somewhat longer. These copper punchings are spaced at intervals with the steel punchings giving, in effect, circular fins projecting beyond the stator iron, see illustration, left. Because these disks are in intimate contact with the iron, where the heat is generated, they are able to conduct heat to the back of the stator where it can be removed by the air recirculated by the fan.



ASSETS to a BOOKCASE

Practical Design of Exhaust and Blower Systems

By E. A. Favalora; published by Edwin A. Scott Publishing Co., Inc., New York; 274 pages, 6 x 8 inches, clothbound; available through Machine Design, $3.00 *postpaid.*

Treating a subject of increasing importance in the machine design field,the author has drawn upon twenty-five years of experience in exhaust and blower system work to present a comprehensive treatment. Avoiding too mathematical an approach, Mr Favalora has introduced numerous examples of typical problems, as well as appropriate charts and drawings. Design of ducts and piping, dampers and hoods, separators and arresters are treated as well as selection of proper fans and blowers and the removal of dust from machines.

The book may be considered to consist of two sections. The first treats fluid flow. Approaching the problem from the emperical aspect, a discussion of individual flow conditions is made, followed by a series of typical problems. Among the subjects considered are: velocity and pressure calculations, test instruments and their use, pipe friction loss, orifice loss, and velocity requirements of various collection systems. The last section of the book which may be defined as covering mechanical and structural design, considers: Blower characteristics, dust separaters, dust removal, framing and other topics.

Of particular interest to designers, illustrations of proper duct and valve designs are given, accompanied by cases of faulty construction. Profuse use is made of time-saving tables and charts.

□ □ □

Plastics Mold Engineering

By J. H. DuBois, executive engineer, Shaw Insulator Co., and W. I. Primble, product engineer, plastic division, General Electric Co.; published by the American Technical Society, Chicago; 494 pages, 6 x 8½-inches, clothbound; available through Machine Design, $7.00 *postpaid.*

Prepared with the intention of discussing the design and use of important types of plastic molds, this book helps to fill a void in design information in a growing field. As is pointed out by the authors, mold-making practice has been considered a trade secret in many quarters. This lack of dissemination of knowledge and experience has retarded the growth of the plastics industry. Accordingly, the book may be considered a timely reference text for the mold designer, but it is also an instructive work of interest to the engineer concerned with the design of plastics parts. This fact is made evident by a consideration of the wide selection of materials available, of the methods of molding from which to choose, and of the design factors contingent upon the molding technique.

Three general aspects of the molding problems are covered in the book. In the first portion, the authors consider the properties of plastics and the proper design of the product. A description is made of the general molding problem, clarifying the problems involved. The more common plastic materials are considered, their physical properties and the types of molding procedure to which they are suited are discussed briefly. Proper design of plastic parts is covered at length. Such factors as dimensional tolerances, parting lines, and the proper use of inserts are given attention. The authors then introduce the reader to the basis of mold design. General types of molds, the applications of these types, the materials for molds, and the manufacturing machines and techniques are covered. The last 300 pages of the book are concerned with the mathematics and geometry of mold design.

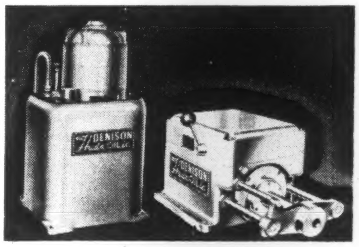
To the engineer concerned with the design of molds, this book will prove to be an excellent source of basic data. The subject is covered in not only a qualitative manner, but in a detailed quantitative way. In a section entitled "Designing and Drafting Room Practice," the authors have compiled numerous specific examples of mold designs in which dimensions and tolerances are given and explained. Under the subject of mold design, examples are given of typical design calculations. A 39 page appendix contains numerous conversion charts and tables of data not commonly available.

An outstanding handbook on leather and synthetic rubber mechanical packings has been published under the title, "Mechanical Packings Manual", by the Graton and Knight Co. Material contained in this volume compiled by Everett H. Clark, manager of the Packings Sales Division of the company, is based upon engineering and research data accumulated over a long period of time in the solving of a great many industrial packings problems.

Replete with drawings, photographs and data sheets, the manual is an excellent textbook for anyone desirous of acquiring a working knowledge of good packing practice. Chapters 3, 4, 5, 6, and 7 should be of considerable assistance to the machine designer in the selection and application of the proper type of mechanical packing as well as in the design of the machine parts adjacent to the packing points on hydraulic or pneumatic equipment. The book, bound in genuine leather, contains 336 pages, 6 by 9 inches, and is available at $4.50 from Graton and Knight Co., Worcester 4, Mass.

new parts and materials

Hydraulic Power-Unit



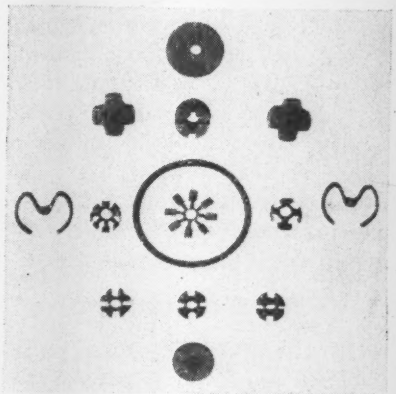
MULTI-UNIT HydrOILic Power—a group of "packaged" hydraulic press components in compact, unit form, has been announced by The Denison Engineering Co., 1160 Dublin Rd., Columbus, O. Fully standardized, a means is offered of installing oil-hydraulic pressure action in equipment especially designed to meet unusual or specific manufacturing needs. These individual components include compact and easily moved pumping units in two models, and five models of power heads. The pumping unit of these multiunits incorporates electric motor, hydraulic pump, pressure regulating valve, reservoir for the hydraulic fluid, and oil level gage. The components are available in 4 and 6 ton capacities, both having the same dimensional specifications.

Precision Snap-Switch

SNAP SWITCHES, identified as class 9007 Type A, have been announced by the Square D Co., 4041 North Richards Street, Milwaukee 12. Measuring 2½ in. long, 1-in. high, and ⅞-in. wide, the snap switch features husky construction, unusual mounting convenience and 550-volt rating. The device is described as a self-contained mechanism for making or breaking a circuit, in a small space, with a short movement and light pressure.

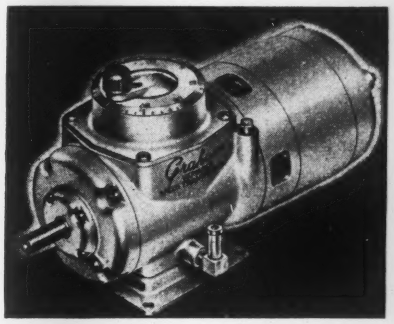
Permanent-Magnet Materials

FIVE NEW permanent-magnet materials, known as Cunico, Cunife, Vectolite, Alnico 12 and Silmanal, have been announced by the General Electric Co., Schenectady, N. Y. Cunico, which is age hardening, is available in rod, strip or wire form, and is furnished only in the final shape. The material, being malleable and machinable, permits screw-machine manufacture. Cunife has the physical properties of Cunico, differing in having directional properties and must be magnetized in the direction of cold-working to obtain highest magnetic quality. The magnets are made from wire stock in round, square or rectangular form. Wire, can, however, be flattened to make thin narrow shapes. Vectolite is the first nonmetallic, nonconducting magnet material made. It is a hardened, sintered combination of iron rust and cobalt oxide. These magnets are extremely light and their nonconductive properties prevent electrical losses caused by conduction of current. Alnico 12 material is cast and must be ground to shape. Magnets of this material are useful in tachometer generators and electronic devices. Silmanal is ductile and malleable and is available in rod, strip or wire form.



Variable-Speed Transmission

NEW MODELS of its variable-speed transmission have been announced by the Graham Transmissions Inc., 3754 N. Holton St., Milwaukee 12. Though bearing no major design change, the models have been improved in detail and materials to provide weight and size reduction as well as improved performance. It is said by the manufacturer that by reasons of manufacturing economies, the new units having the same standards as the old are available at



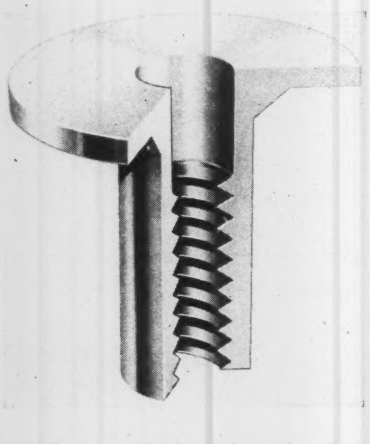
prices in line with low-cost equipment. Two sizes are offered: Model 15 for motors up to ⅛-hp, and model 40 for motors to 3/4-hp. Both models are available in a variety of controls.

Plug-In Relays

HERMETICALLY sealed plug-in relays developed by the Ward Leonard Electric Co., Mount Vernon, N. Y., are designed for application where space is limited. Completely encased in a cylindrical can, these compact midget type relays provide protection against adverse atmospheric conditions such as moisture, dust, gases, corrosion, etc. Relay coil and contact connections are totally enclosed within the metal housing and are brought to the prongs of a standard octal plug base. Vibration resistant, the relays are available in contact combinations up to double-pole double-throw, with a-c contact ratings of 4 amperes from 0 to 115 volts, and d-c contact ratings of 0.5 amperes from 25 to 115 volts.

New Fastener

WASHER HEAD RIVNUTS, one piece internally-threaded and counterbored tubular rivets, have been developed for use in fastening plywood sections to steel supports. Announced by the B. F. Goodrich Co., Akron, the Rivnut at present available for this purpose is cadmium-plated steel in 6-32 thread. The fastener may be used for fastening ¼-inch plywood to 1/16-inch steel sections. It is said of the Rivnut that it will not vibrate loose, will not pull through the plywood, and that it fits flush with the surface.

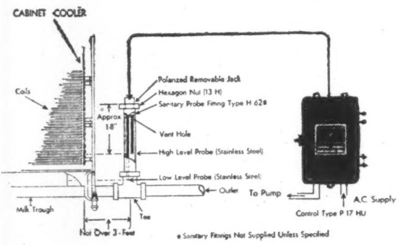


Small Snap Switch

DESIGNED AND marketed by Robert Hetherington & Son Inc., Sharon Hill, Pa., the new small snap switch is only 9/16-in. in diameter and 31/32-in. in length overall. Complete in itself, it can also be furnished with special over-travel adapters and special mounting sockets. A pressure of 4 lb is required to operate this switch which is rated at 15 amp at 110 v alternating current. It can be supplied normally-open, normally-closed, two-circuits or single-pole double-throw. Load ratings are based on the tungsten filament test.

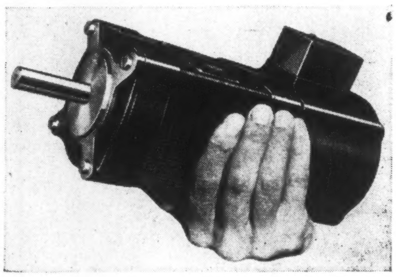
Sanitary Level Control

SANITARY LEVEL control series P17, offered by Photoswitch Inc., 77 Broadway, Cambridge 42, Mass., makes available to the dairy and food processing industries a series of electronic relays for level control of all liquids. Contact with the liquid is made only by stainless steel probe rods. Being a floatless level control, containing no vacuum tubes and operating with low voltage in the probe circuit, Series P17 meets all requirements of sanitary and health boards. Stainless steel probe rods are suspended into the tank from the probe fittings. When level of liquid in the tank falls below lower probe, level control closes the circuit controlling the pump and the tank fills. When liquid rises to the level of the upper probe, the circuit controlling the pump opens and the pumping operation stops. For pumping-out control, operations are reversed. The starter is designed for 115 or 230 volts circuit, 60 cycles alternating current. Power consumption is ½-watt. Contacts are 300 watts at 115 or 230 volts alternating current. The probe circuit will carry a resistance as high as 3500 ohms.

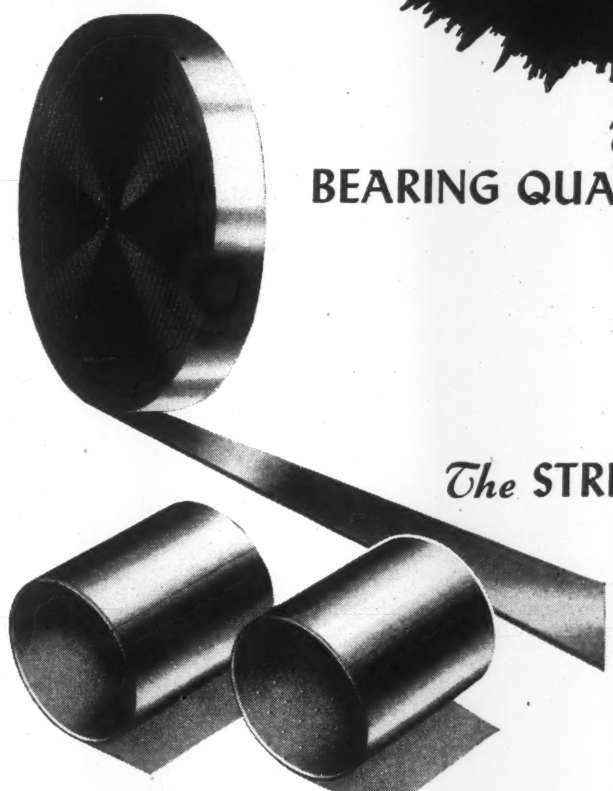


Miniature 2 hp Motor

AN EXTRA-COMPACT and light 2 hp explosionproof motor has been developed for continuous duty by Electrical Engineering and Mfg. Corp., 4606 W. Jefferson Blvd., Los Angeles. A direct current aircraft type with armature speed of 9000 rpm, the motor is available at any output shaft speed desired. An integrally built gear reducer, supplied where lower speeds are required, adds only two pounds to the basic weight of 15 pounds. A ball bearing motor,

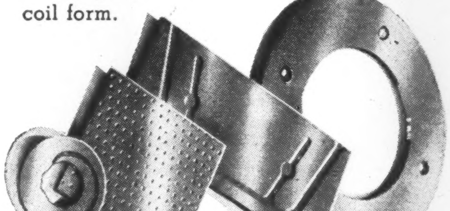


Something NEW in Bearings



The BEARING QUALITIES of Pre-Cast Bearing Bronze on The STRENGTH of PATENTED Steel

Made to your precise specification in Sleeve Bearings, fabricated into washers, thrust plates and sold in coil form.



. . . Write TODAY for this new literature on Johnson BRONZE-ON-STEEL.

Pre-Cast Bearing
Bronze
on Steel
SOMETHING NEW in BEARINGS
by JOHNSON BRONZE

Here is a new and unusual Sleeve Bearing material that combines the excellent bearing characteristics of a high grade bronze alloy with the strength of steel. It enables manufacturers to increase speeds and loads . . . to give longer bearing life with smooth operation . . . plus greater resistance to shock and to wear. Pre-Cast Bearing BRONZE-ON-STEEL produces a thin wall, laminated type of bearing. The fabricating process, essentially a series of stamping and forming operation, provides accuracy and precision. It is low in cost, particularly in long production runs.

In addition to its wide use as Sleeve Bearings, Pre-Cast BRONZE-ON-STEEL serves many other important applications in industry. Many concerns use it for stampings, washers, guide strips and other flat pieces. For such purposes we can supply it in coils with a maximum width of 5-1/2" and a range of thickness from 1/32" to 3/32".

Excellent delivery is now possible on both finished bearings, parts or coils. Why not investigate the possibilities of using BRONZE-ON-STEEL in your product? A Johnson Engineer will be glad to show you how and to explain its many advantages. There is a Johnson Bronze man as near as your phone. Why not call him in TODAY?

JOHNSON BRONZE

SLEEVE BEARING HEADQUARTERS

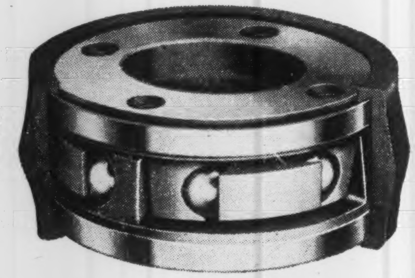
The MOST COMPLETE SLEEVE BEARING SERVICE in the WORLD

525 S. MILL STREET NEW CASTLE, PA.

it is completely insulated with glass insulation and high temperature varnishes, operating temperature is rated at 75 C over an ambient of 25 C. Models are available for operation on 28, 32, or 110 volts, direct current.

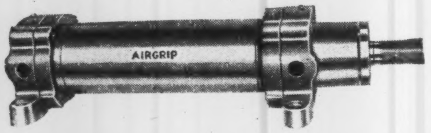
Flexible Coupling

INTRODUCED by the Gear Grinding Machine Co., 3901 Christopher St., Detroit 11, is a ball type of axial and angular flexible coupling. Named the "Ballflex", the coupling obtains its flexibility by mechanical means only. Friction is reduced by the use of hardened and ground steel balls through which all torque is transmitted. Free rolling in any direction, side thrust and bending motion, both in the coupling and mounting bearing is eliminated. The unit is made without flexible disks or sliding members. The dust seal is an oil impervious synthetic material which retains the lubricant. Available for shafts to 1 3/4-inches in diameter, the coupling will accommodate parallel misalignment up to 1/8-inch, and angularity up to 3 degrees. Capacity ratings are three horsepower per 1000 rpm for normal loads, to one horsepower per 100 rpm for heavy shock loads.



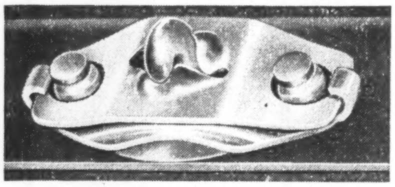
Hydraulic Cylinders

ANNOUNCED by the Anker-Holth Mfg. Co., Port Huron, Mich., is a complete new line of non-rotating hydraulic cylinders. Designed for pressures to 2000 psi, the cylinders include special features such as: Cushioned cylinders, two-way action, step-seal piston rings, and safety factor of 6 to 1. The cylinders are available in a number of standard mountings, and a wide range of strokes and sizes are furnished.



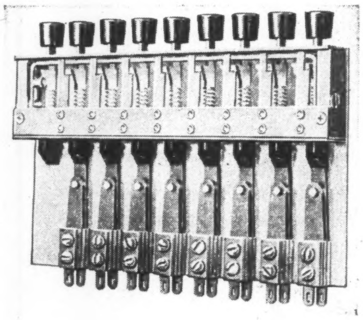
New Fastener

IN USE DURING the war on combat planes, the Lion fastener is now available for peacetime application. Manufactured by the Lion Fastener, Inc., Honeoye Falls, N. Y., it is a positive locking device to be applied to removable plywood, metal, plastic strips, channels or housings. Consisting of two parts—stud and cantilever spring assembly—the stud is of solid one-piece construction with shoulders forged in opposite sides to form the locking member. Studs are available in headless, flush head, or wing head types. The spring assembly, comprised of cantilever type spring and base, is a pre-assembled unit ready to install. The base forms a supporting member for the spring under extreme loads. Of floating leaf construction, the spring is held by retained lugs at each end of the base. It is formed with a double set of grooves and cams which firmly interlock with the notched shoulders of the stud. Spot welded or riveted in place, the fasteners eliminate projecting handles, knobs and heavy bolts.

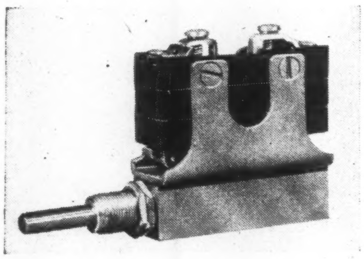


Multiple-Position Switch

ANNOUNCED by the General Control Co., 1200 Soldiers Field Rd., Boston 34, is a new nine-position pushbutton switch. The switch is of sturdy construction with all parts of noncorrosive materials. Contacts are fine silver, permanently riveted to phosphor-bronze springs. The unit is available also in a locking frame type which has eight positions and one reset position. In this type any switching combination which has been set can be released by one operation of the reset button.

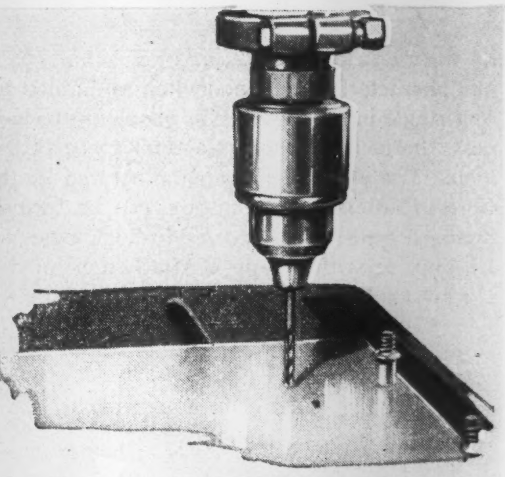


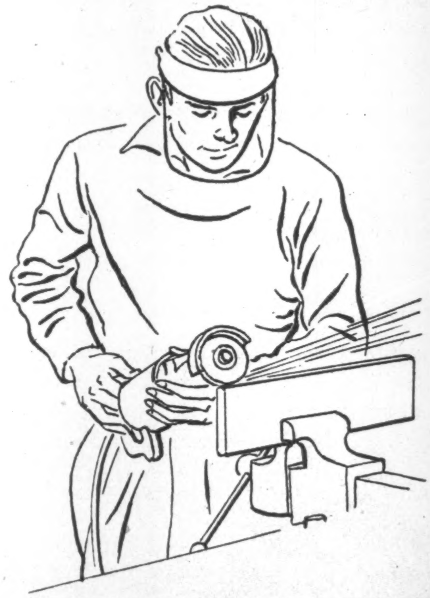
Switch with Over-Travel Device



DESIGNED to provide flexibility in applications where the switch may be subjected to shock and vibration, the type EP-203 Mu-Switch assembly comprises a standard Mu-Switch contained in a rigid bracket in which is incorporated a long-throw over-travel mechanism. The operating force is applied in a direction parallel to the moving elements of the switch rather than perpendicular as has been customary in the past. The force and movement characteristics are governed

How magnesium is speeding production because it's easy to work!





Many a firm goes to magnesium for its unique *lightness . . .* and then chalks up additional boosts in production because it's so *easy to work*. Modern plants that use magnesium find it saves time, labor, and tools through this exceptional workability.

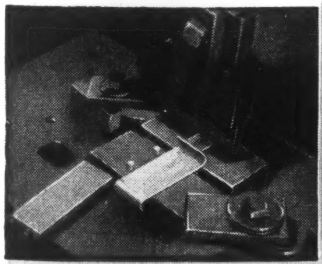
Magnesium is easy to saw, with band and circular saws and also hand and power hack saws. It permits larger cuts per tooth than other structural metals. And it can be worked faster and easier than most metals by hand tools—chipping tools, drills, burrs, chisels, planes, and portable milling cutters.

As the first step toward applying these important economies to your own product, you need only get in touch with the nearest Dow office. Technical consultants are at your service.

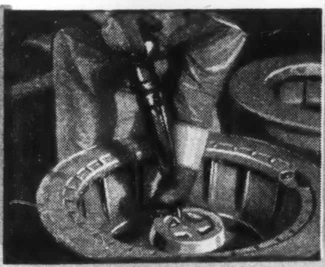
Ready... to make products move!

MAGNESIUM

LIGHTEST OF ALL STRUCTURAL METALS



Magnesium forming methods are similar to those used with other metals, except that magnesium is usually formed hot.



Excellent machinability of magnesium speeds work with hand tools, such as the chipping operation on this magnesium casting.

Sawing magnesium, with low cutting pressures, requires only one-tenth the power that is needed for sawing steel.

DOW

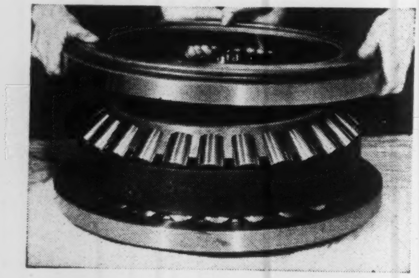
DOWMETAL

MAGNESIUM DIVISION • THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN
New York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

by the overtravel device, and are independent of the normal characteristics of the switch assembled to it. Permitting single-hole mounting in panels up to ½-inch in thickness, the switch occupies a total space of 2¼ x 2¼ x ⅞-inch. The unit, which is manufactured by the Mu-Switch Corp., Canton, Massachusetts, can be furnished with the standard type M or type A Mu-Switch for ac loads up to 15 amp. or with a type F Mu-Switch for dc loads up to 1¼ kw at 230 v.

Spherical Roller Thrust Bearing

DEVELOPMENT of a new type of roller bearing capable of carrying heavier loads at higher speeds and lower temperatures is announced by SKF Industries, Inc., Front St. and Erie Ave., Philadelphia 34. The bearing's self-aligning principle, which compensates for shaft deflections, distortions or weaves, permits heavy loads to be distributed evenly over all rollers and eliminates danger of overloading. Ability of the bearing to carry heavy loads at high speeds is said to be possible because of unique design permitting more effective lubrication. Making a contained assembly, a cage retaining sleeve pressed into the bore of the inner ring contains the rollers, cage and inner ring. Typical applications of the new bearing include marine propeller shafts blocks, roll necks, thrust mountings for railroad turntables, water wheels, and extrusion machinery.

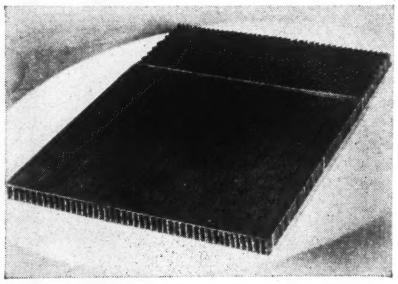


Sensitive Electronic Relay

FUNCTIONING by means of an application of the hot-cathode thyratron, a new electronic relay has been announced by the Automatic Temperature Control Co., 34 E. Logan St., Philadelphia 44. Operating from a 60-cycle 110-volt power supply, the switch is snap acting, and has a high-load contact capacity. It trips by either an external contact carrying three microamperes or any external circuit the value of which drops to one megohm pure resistance. Applications suggested are: Use in phototube circuits, motion and limit control, and liquid-level control.

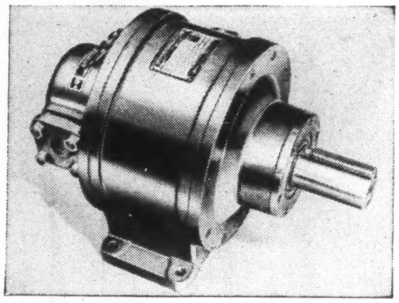
Lightweight Structural Material

FEATHERWEIGHT construction material of high strength has been announced jointly by the Glenn L. Martin Co. of Baltimore, and the United States Plywood Corp. 616 W. 46 St. New York. Made of a newly developed "honeycomb" of phenolic impregnated cloth or paper sandwiched between and firmly bonded to thin sheets of facing material, the new waterproof sheets are said to possess high structural strength. Various materials such as aluminum alloys, stainless steel, wood veneers, or plastics can be used for facings either singly or in combination. The edges of the honeycomb sheets may be sealed against moisture, making slab construction feasible without fear of warping. Thickness of the core may be from ⅛-inch upward, and sheets as large as 7 x 30 feet have been made. Curved sheets are under development.

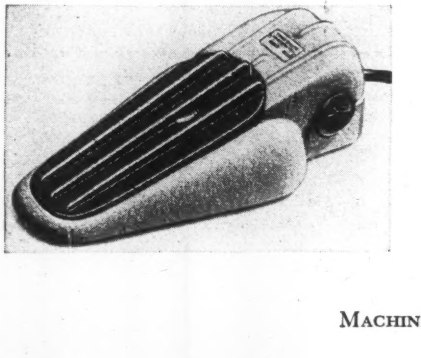


Hydraulic Pump

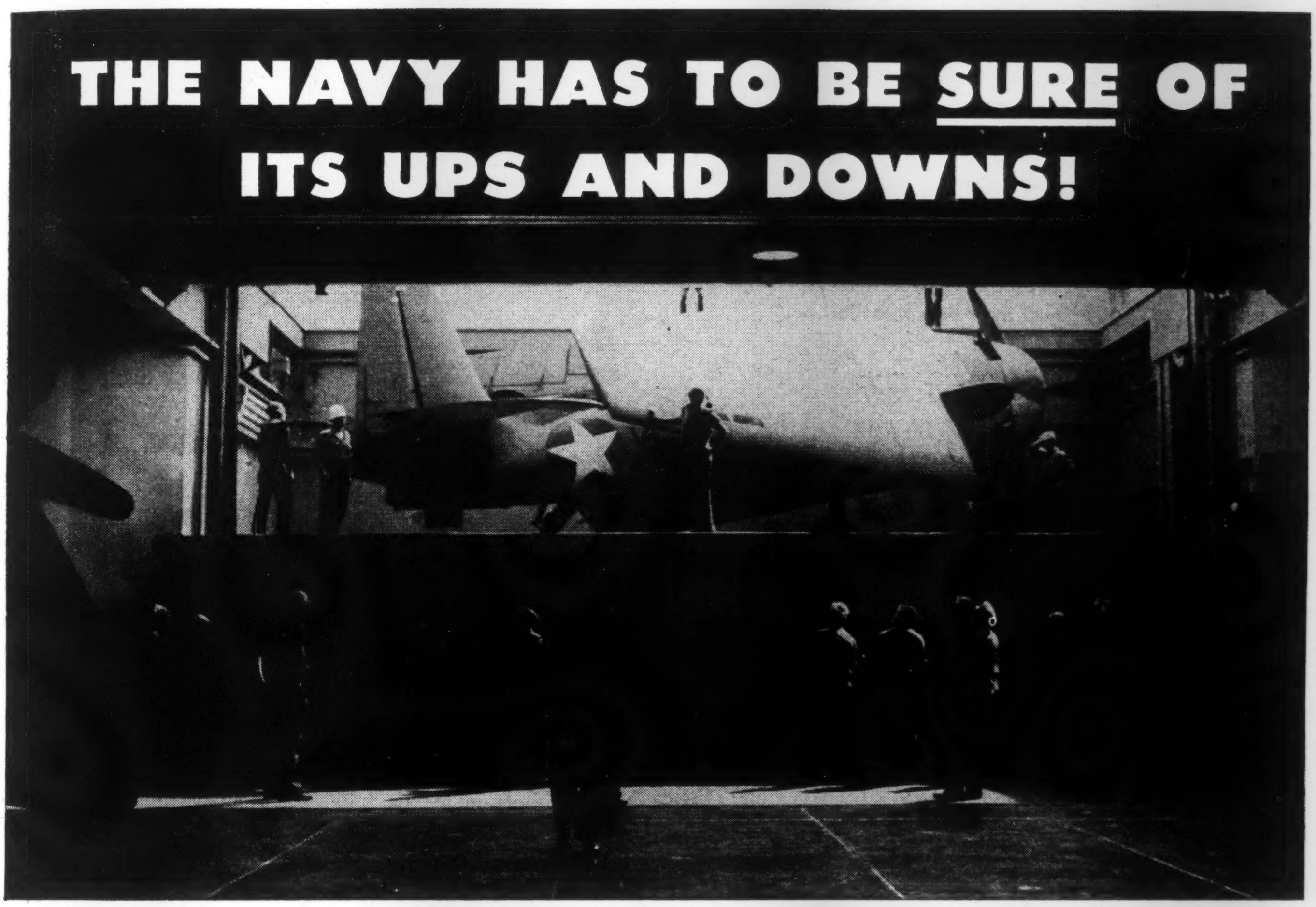
PROVIDING a compact source of hydraulic power, a 5000-lb unit called the "Superdraulic Pump", is announced (see MACHINE DESIGN, March, P. 141) by Hydraulic Machinery, Inc., and Superdraulic Corp., 12825 Ford Rd., Dearborn Mich. Utilizing multiple, radial pistons driven by female, elliptical cams, the pump is available in either constant or variable delivery types. The unit is claimed by the manufacturer to eliminate the need for intensifiers and accumulators, and to provide a single source for high-pressure, high-volume hydraulic power. Light enough for a man to carry, the pump is rated at 5000 psi at 40 hp and 1200 rpm. Capacity may be varied from zero to seventeen gallons per minute.



Double-Throw Foot Switch



PRODUCTION has recently been announced by the Industrial Timer Corp., 117 Edison Place, Newark 5, N. J., of the "Time-O-Light" foot switch. Incorporating a single-



THE NAVY HAS TO BE SURE OF ITS UPS AND DOWNS!

How VIM Leather Packings solved a wartime problem

. . . and serve in peace as well

A VIM Leather resin-impregnated "V" packing, supplied in sets for hydraulic use. Houghton also furnishes synthetic rubber packings and "O" Rings.

When cruisers were being hurriedly converted to "flat-tops" to give the Navy more eyes for submarine detection, a serious problem arose. Hydraulic elevators failed to function perfectly; planes on storage deck were delayed getting "upstairs," and those on the flight deck couldn't be stowed away rapidly enough to make room for more home-coming "birds."

The shipbuilder, elevator supplier and Navy officials conferred . . . called in the Houghton Man. A simple change in packing design was recommended and made, and the hydraulic mechanism worked perfectly thereafter.

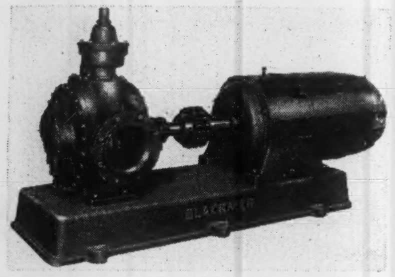
The war is over, but need for packings increases as peace-time hydraulic applications multiply. Our technical staff is ready to help in *your* design problem. Write E. F. HOUGHTON & CO., 303 W. Lehigh Ave., Philadelphia 33, Pa., and all principal cities.

HOUGHTON'S *Engineered* VIM *Leather Packings*

pole, double-throw, snap-action switch having silver contacts, it is sturdily built, with non-skid base. When the treadle is depressed, 1500 watts is supplied to each of two outlets. With the treadle in the normal position, 1500 watts is supplied to a third outlet only. A low-power neon light in the pedal indicates its position in the dark. The switch may be used with photographic equipment, in which connection the pilot light is not strong enough to fog photographic paper. Motor-driven industrial equipment up to one horsepower also may be controlled.

Pump Unit For Oil

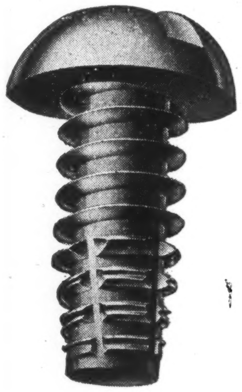
FOR HANDLING bunker "C" fuel oil and liquids of similar characteristics, a new special rotary pumping unit has been developed by Blackmer Pump Co., Grand Rapids, Mich. Operating on the "bucket design" (swinging vane) principle, the pump is self-adjusting for wear, with no loss of capacity during the life of the buckets. When worn to the critical point these may be replaced by removing the head of the pump, pulling out the worn ones, and inserting new buckets. Power unit of the pump is a gearhead motor of 50 hp with a speed of 150 rpm at the drive shaft connected to the pump by a flexible coupling. Pump has a capacity of 500 gpm, and operates at a pressure of 125 psi.



Remotely Operated Switch

BEING FREQUENCY-CONTROLLED tuned relays, the switches developed by Stevens-Arnold Co. Inc., 22 Elkins St., South Boston, Mass., are particularly well suited for remote control applications either with wire or radio as the carrier medium. For radio application the switches are sufficiently sensitive so that when made on special order that they can be controlled from either a crystal detector or vacuum-tube receiver. Response time is only a small fraction of a second. Switches are adjusted to accept a selected band of frequencies in the range of 20 to 800 cycles per second and to reject all others. More than one switch may be connected together, providing a means of selective switching by a choice of frequency bands. Ratings of 1 to 10 amperes, 115 volts alternating current are available.

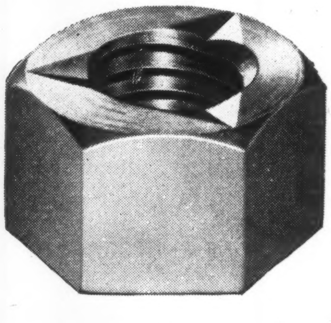
Self-Tapping Screw



DESIGNED especially for making fastenings to comparatively thin sections and bosses in friable and brittle plastics, the type F-Z self-tapping screw has been announced by Parker-Kalon, 200 Varick St., New York 14, N. Y. This screw combines the thread-cutting characteristics of the P-K type F screw with the coarse-pitch thread advantages of the P-K Type Z screw. Five cutting flutes of this fastener distribute cutting pressure evenly, prevent localized pressure and cracking, especially in brittle plastics, by permitting chips to drop to the bottom of the hole. The low stresses said to be set up by this new screw permit design of plastic shells with more uniform wall sections eliminating the necessity of inserts in plastic assemblies with no sacrifice in strength.

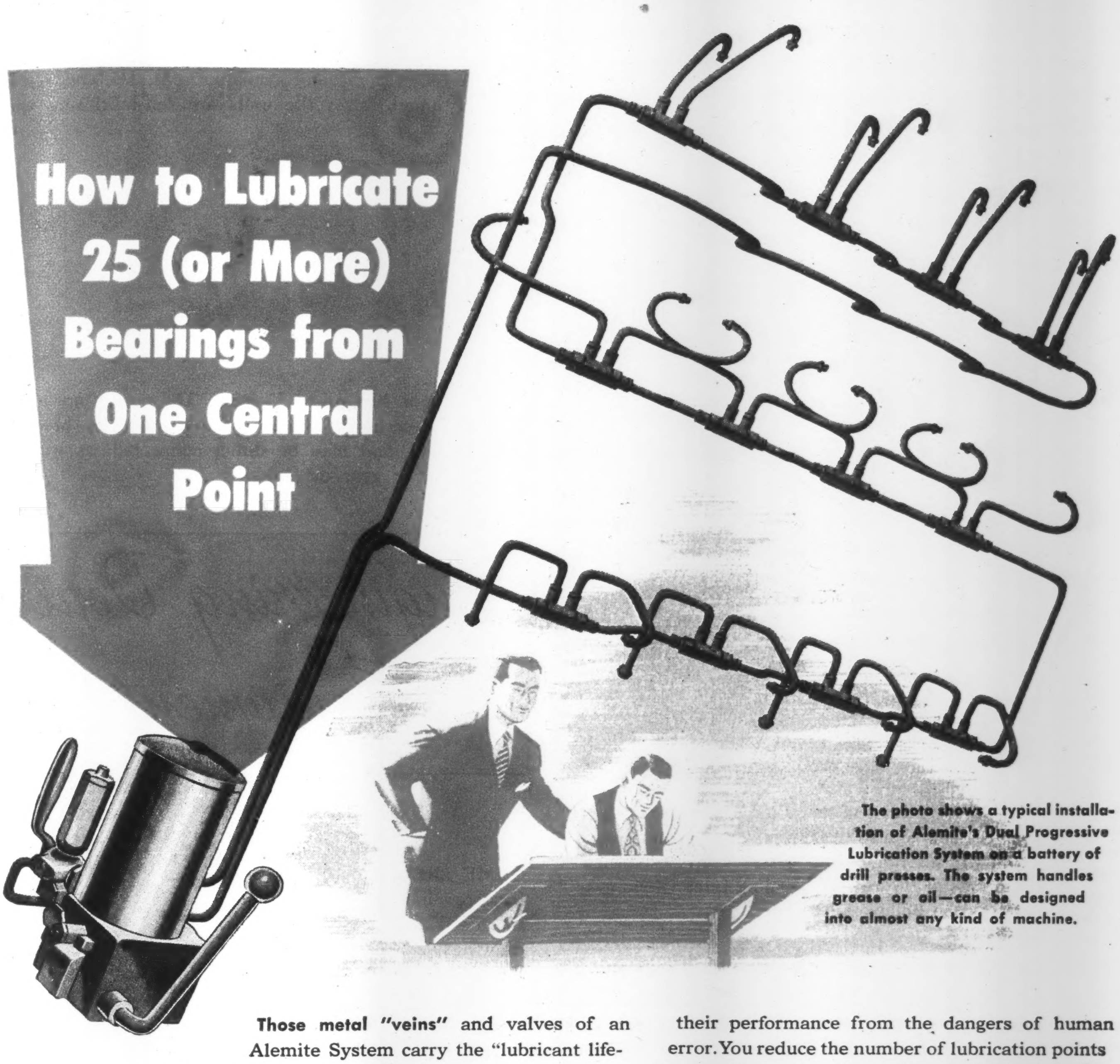
One-Piece Lock Nuts

SEMIFINISHED, one-piece, self-locking nuts of standard dimensions, offered by Grip Nut Co., 310-W South Michigan Ave., Chicago 4, are not affected by oil, water or chemicals. The triangular deflections on top of the nut provide a controlled friction lock. Nut is free-spinning until the bolt threads contact the deflected threads near the top of the nut. These hexagon nuts are available in the following sizes: 1/4, 5/16, 3/8, 7/16, 1/2, 9/16, 5/8, and 3/4-inch (American Standard heavy and American Standard regular).



New Plastic

AN INDUSTRIAL plastic that withstands acids which dissolve gold and platinum and retains its strength and form through wide temperature ranges has been announced by E. I. duPont de Nemours and Co., Wilming-



How to Lubricate 25 (or More) Bearings from One Central Point

The photo shows a typical installation of Alemite's Dual Progressive Lubrication System on a battery of drill presses. The system handles grease or oil—can be designed into almost any kind of machine.

Those metal "veins" and valves of an Alemite System carry the "lubricant lifeblood" to every bearing surface on a machine. Think of it in terms of the machine—any machine—you're designing now.

No matter if your machine or a battery of them has 10, or 25, or 40, 60, or 75 lubrication points. The Alemite System reduces them all to *one*—the one central point where the lubricant is introduced. From there, each bearing receives a measured quantity of lubricant—*and without stopping the machines.*

Proof? For example: a battery of 7 big automatic punch presses has a total of 86 bearings. It took 20 minutes to lubricate each press—and it had to be shut down. Bearing failures were frequent. Now all 86 bearings are lubricated from one central point *while the presses are in operation.* Bearing failures due to faulty lubrication have ended.

When you design Alemite Centralized Lubrication into your machines you automatically protect their performance from the dangers of human error. You reduce the number of lubrication points to *one.* You guarantee positive lubrication without "time-outs." You cut repairs to a minimum . . . lengthen machine life . . . increase customer satisfaction. It all adds up to More Productive Time per machine. And that enhances your reputation as a designer.

* * *

Without obligation, have an Alemite Lubrication Specialist demonstrate one or all 4 new Centralized Systems right at your desk with transparent working models. Also, ask for any technical help you want regarding Alemite Systems. Drop a note on your letterhead to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

ALEMITE

REG. U. S. PAT. OFF.

Another Product of STEWART WARNER

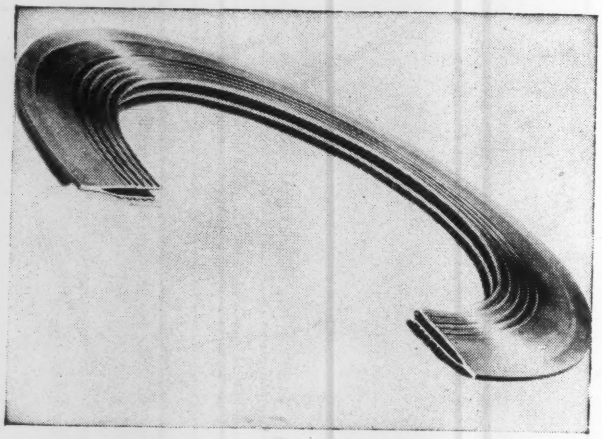
Alemite ALONE *Combines all 3 in Lubrication*

1. EQUIPMENT 2. PROCEDURES 3. LUBRICANTS

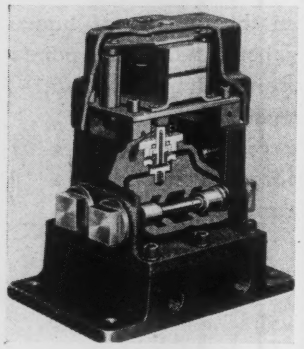
ton Del. Known as "Teflon", the plastic will withstand strong acids and alkalies, and suffers no degradation or adversely affected physical properties at either high, or very low temperatures. Other desirable properties are low power factor, high impact strength, low water absorption, and toughness.

Pressure-Sealed Gasket

OF SERRATED TYPE, a new gasket of Goetze Gasket & Packing Co. Inc., New Brunswick, N. J., consists of two disks of metal machined on their external faces with standard serrations and welded together around their outer periphery. The new gasket combines the pressure and corrosion-resistant qualities of metal gaskets with the light bolting requirements of a soft sealing medium. Line pressure entering the interior of the gasket exerts expansion pressure in excess of the required sealing force. The gasket is known as the Bellowseal.



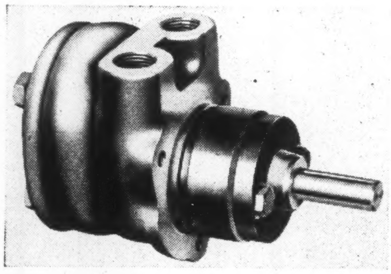
Solenoid-Controlled Air Valve



RECENTLY announced by the Ross Operating Valve Co., 6485 Epworth Boulevard, Detroit 10, is a piston-poppet solenoid-controlled pneumatic valve. Said to allow operating speeds up to 400 cycles per minute, with current consumption of 1.2 amperes at 110 volts 60 cycles, the unit is made with interchangeable parts allowing piston poppet replacement with small labor. A model is available in 3-way normally-open or normally-closed, 4-way, and in ¼, ⅜, ½, and ¾-inch sizes.

Rotary Pump

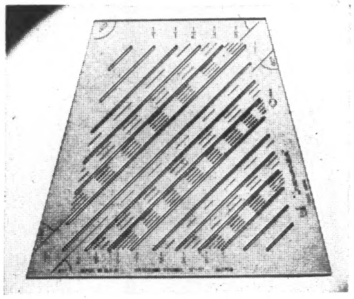
MANUFACTURED by the Geo. D. Roper Corp., Rockford, Illinois, a new series of rotary pumps is useful for both hydraulic and pumping service. The units are recommended for diesel fuel-oil transfer, pressure lubrication, hydraulic service, fuel supply, etc. Known as the Series K rotary pump and available in ¾, 1½, and 3 gallons per minute sizes the units are self-lubricated and handle suction lifts up to 25 feet. The pumps are available with a series of interchangeable mounting brackets and adapters, and may be direct connected, as well as chain, belt, or gear driven.



Engineering Dept Equipment

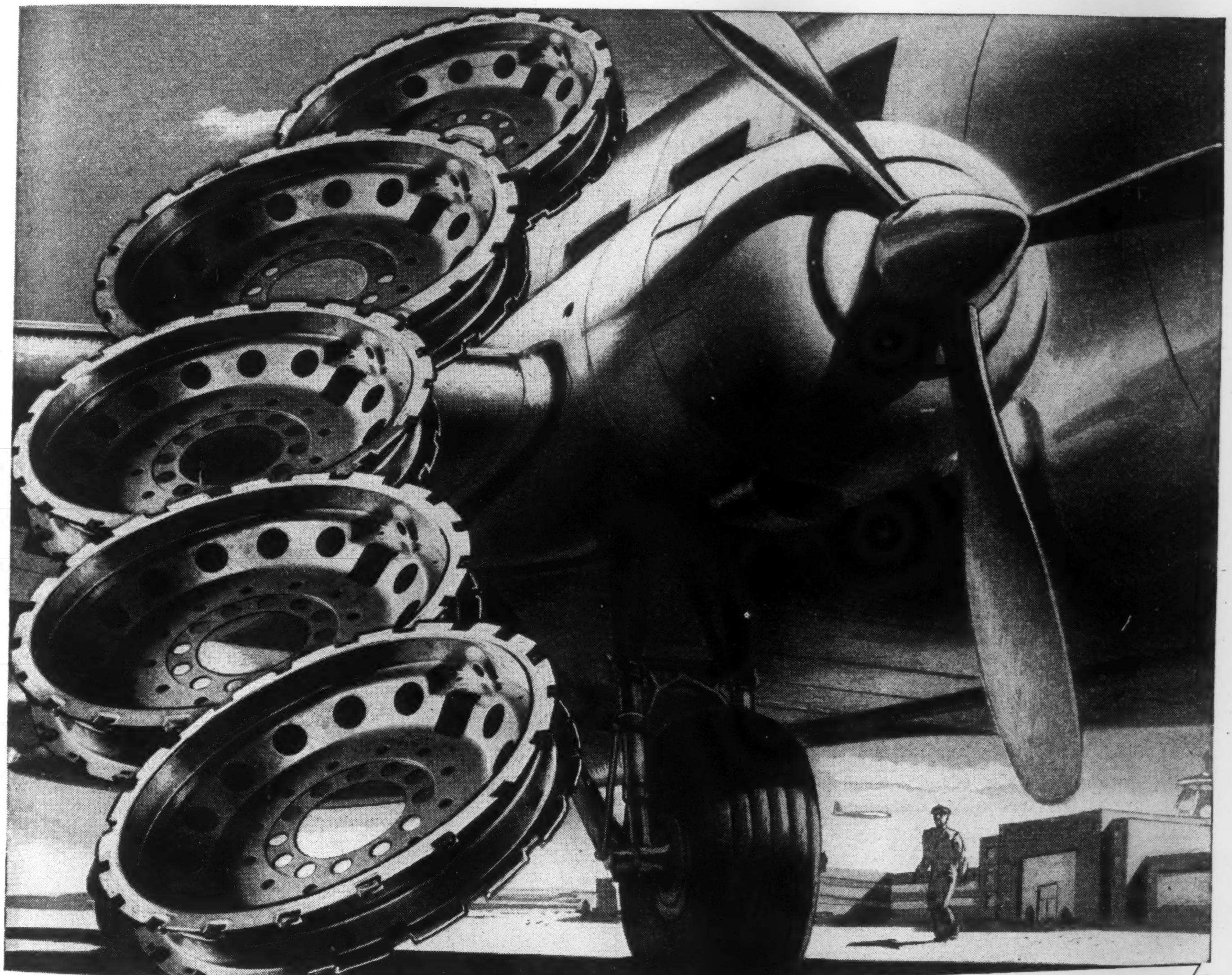
New Hatching Stencils

TIME REQUIRED for cross-hatching is shortened and neater drawings are produced by use of the new Instrumaster hatching stencils announced by Instrumaster Industries, Inc. Distributed by John R. Cassell Co., Inc., 110 W. 42nd Street, New York 18, the stencil is accurately made from clear, nonwarping, nonflammable plastics stock. Aligning marks and lettering are indelible debossed, lines at 30, 45, and 60 degrees to the horizontal can be drawn without a separate triangle. Stencils are available in 5 by 6 inches, and 6 by 8 inches, sizes.



Portable Tachometer

DESIGNED for accurate testing of over-speed protection on turbo electric generators and similar applications, the type 48A sensitive tachometer is manufactured by the Metron Instrument Co., 432 Lincoln St., Denver. Measurements are obtained by use of a low torque head functioning as a positive-acting switch connecting reactance in a bridge circuit. Meter indication is thus directly proportional to the number of switching operations



Brake Wheels for Airplanes
MADE OF J&L OTISCOLOY STEEL

—For Higher Strength Without Increased Weight

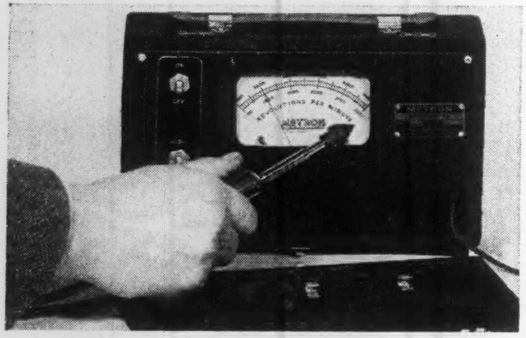
Hundreds of thousands of the combination wheels and brakes shown above were made of J&L Otiscoloy sheets to provide extra strength in supporting weight of practically all big transports, bombers and cargo planes. The 40% higher yield strength of Otiscoloy made it particularly suitable for this job, especially as Otiscoloy has the formability necessary to mass produce this intricate part. Otiscoloy also has excellent weldability and resists corrosion and abrasion. Write for Otiscoloy booklet.

JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PENNSYLVANIA

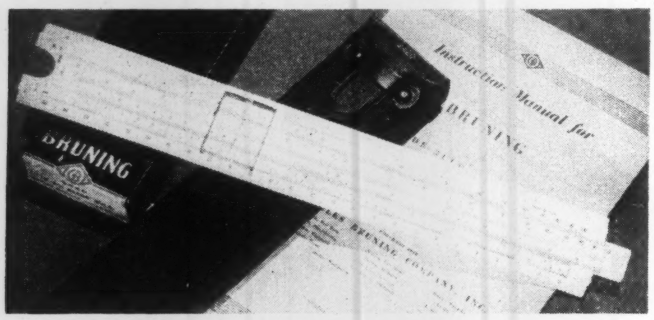
J&L STEEL

per minute. Neither the balance nor the response of the bridge circuit is affected by line voltage fluctuations between 105 and 125 volts in the power source. No vacuum tubes or other limited-life components are incorporated in the design. Having ranges for 1800 and 3600 rpm machines with indicated accuracy within one quarter of one per cent, the instrument is said to have excellent stability against time, use, temperature, and humidity effects. It is supplied complete in carrying case.



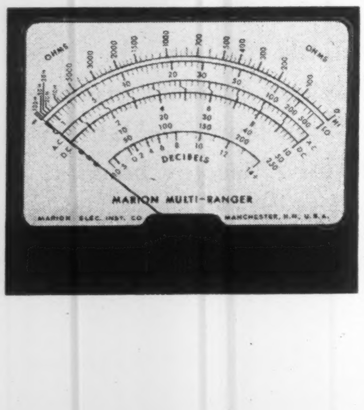
Plastic Sliderule

TO FILL the need for a slide rule of greater dimensional stability, the Charles Bruning Co., 4754 Montrose Ave., Chicago 41, has recently introduced a new 10-inch slide rule. Made of a plastic material said to have remarkable dimensional stability, the precision graduations are not affected by temperature change. The glass indicator is mounted in a polished stainless steel frame that holds it firmly in place. Provided on the rule are: A, B, CI, D, K, S, L and T scales, as well as inch and centimeter scales on the beveled edges of the rule.



Versatile Meter

DESIGNED to permit the user to assemble a highly accurate instrument for use as a voltmeter, milliammeter, high and low resistance ohmmeter, a-c voltmeter or decibel meter, a new instrument has been announced by the Marion Electrical Instrument Co., Manchester, N. H. Known as the Multi-Ranger, the instrument is available in 3½, 4½ and 8½-inch sizes and each size is interchangeable electrically. Scales are printed in three colors for quick identification, covering a 100 degree arc. Ranges normally supplied include 0-10-50-250 d-c volts, 0-10-50-250 a-c volts, 0-500 ohms and −10 megohms, −10 to +14 decibels. By use of the proper multiplier the 0-10 voltmeter scale can also be used as 0-1000 volt scale. Basic sensitivity of the instrument is 400 microamperes, and the internal resistance is 500 ohms. Alnico magnets are used with the Marion "Bulldozer" moving system.



White Plastic Diffusers

FOUR-FOOT WHITE plastic diffusers, announced by Edwin F. Guth Co., 2615 Washington Blvd., St. Louis 3, snap on or off 40-watt fluorescent lamps to reduce lamp brightness and provide efficient diffusion. Known as PFC-100's, they minimize glare. The diffusers have a spring-like design that grips the lamp along its length to afford easy cleaning and relamping. The shields are said to confine lamp heat for better operating temperatures.

Plastics

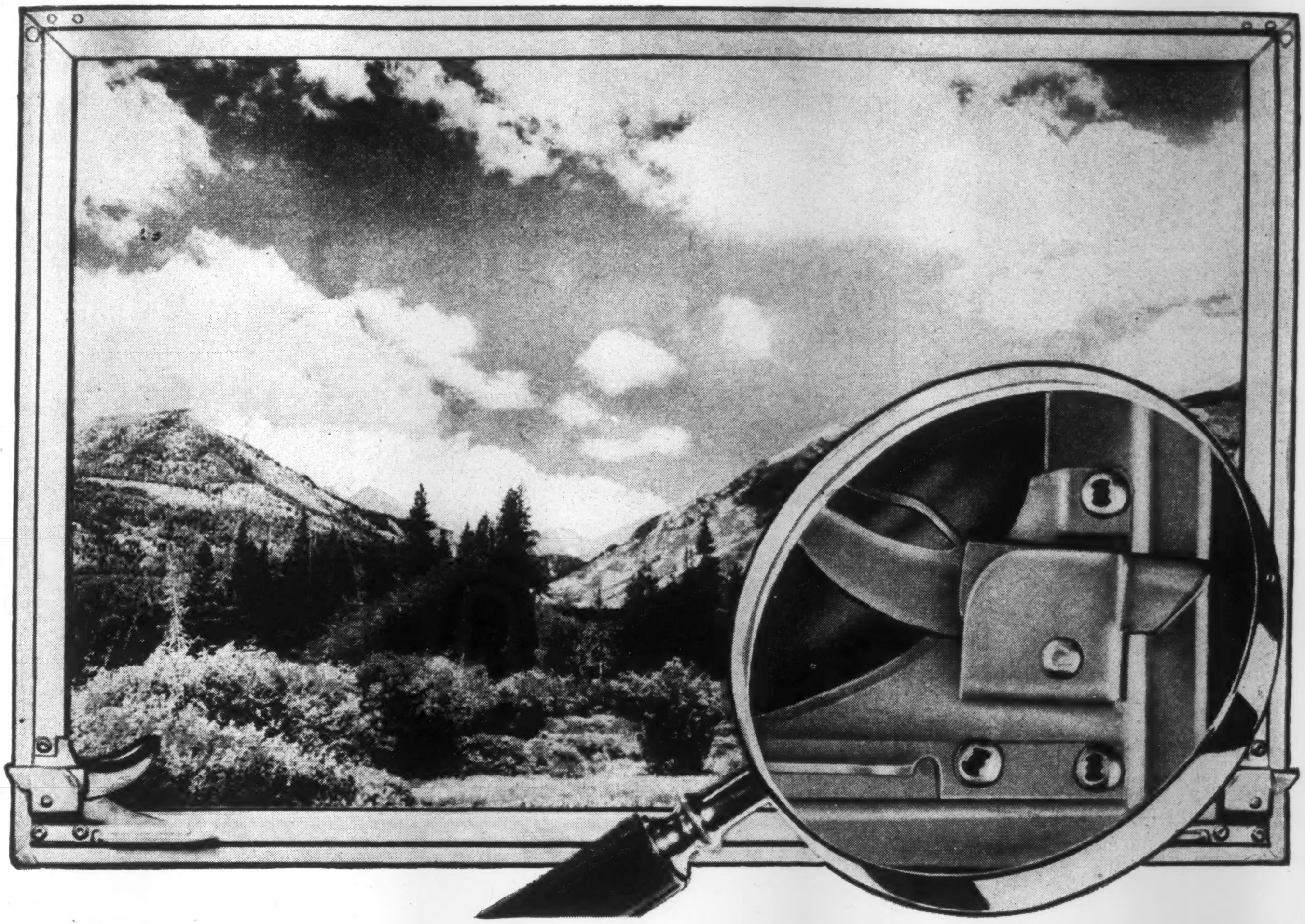
(*Concluded from Page* 118)

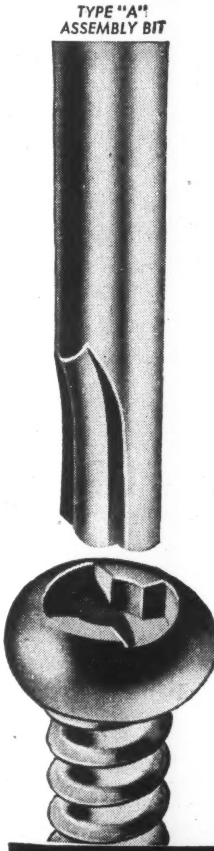
this will do much to insure a better distribution of insulation and eliminate hot-spots which often are caused by air pockets. In addition, a neater and more mechanically secure insulation is obtained. There is, however, a decided problem of minimizing stresses due to differential expansion and contraction between the metal and the plastic. In applying molded plastic insulation, the armature or field coil, complete with windings, is placed into the mold as an insert and the plastic forced in under pressures of 3000 to 5000 psi.

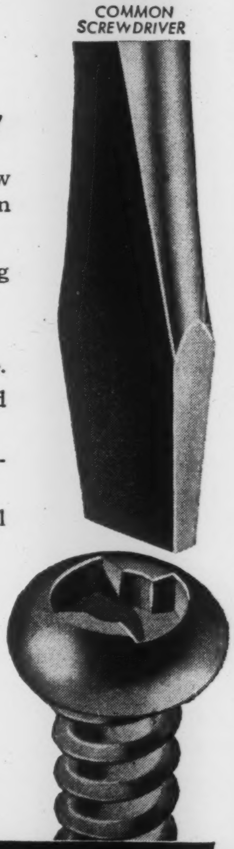
An extensive line of generators and starters was developed by Jack & Heintz Inc. during the war for military and commercial aircraft. One large bomber was equipped with as many as twenty-four of these units. Molded parts are prominently featured throughout the design and construction of these starters and generators. Some of the molded phenolic parts are shown in *Fig.* 6. It is also interesting to note that many of these parts were produced by jet molding methods.

In connection with molded parts produced for motors, there has been a pronounced standardization toward molded brush holders with metal inserts such as that shown in *Fig.* 7. Pressure is usually exerted upon the brushes by means of a spring located under the cap. A brush rocker ring and slip ring are also shown in *Fig.* 7. Insulation outside of the brush and the brush support must be substantial, however, to prevent losses or leakage to the frame of the motor.

Through sound application of basic engineering principles and through correct understanding of the electrical properties of plastics, designers will find it possible to develop new and improved machines offering optimum performance.







Clutch Head Screws

By-pass Skid Damage for Excel Corporation

At this Elkhart, Indiana, plant the manufacture and assembly of steel window sash for Transcontinental, City, School and other types of bus transportation is on a production basis.

Excel Corporation says: "We have found many advantages since standardizing on CLUTCH HEAD Screws:

- We have definitely increased production.
- We have by-passed damage to the painted surface through driver slippage.
- We feel that the surer driving with CLUTCH HEADS is an important safeguard against injury to our operators.
- Simplified operation with an ordinary screwdriver has freed us from complaints regarding field-servicing difficulties.
- The longer life and easy reconditioning of the Type "A" Bit reduces tool cost and saves time."

Send for samples of screws and Type "A" Bit

Non-Tapered Driving

All-square contact of screw and bit eliminates "ride-out" for safer, easier driving. A 60-second application of the end surface to a grinding wheel repeatedly restores this Type "A" Bit to original efficiency.

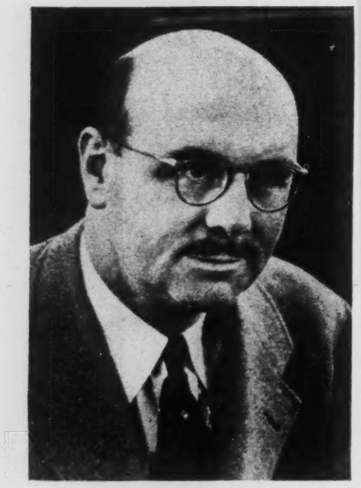
CLUTCH HEAD SCREWS

No Servicing Problems

because CLUTCH HEAD is basically designed to operate with any common screwdriver which need only be reasonably accurate in width. Thickness of the blade is a secondary consideration.

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7



Thomas J. Bannan

Arthur E. Raymond

Harry R. Greenlee

MEN... of... machines

THOMAS J. BANNAN, who has been active in American Gear Manufacturers Association activities for a number of years, serving as vice president for the past year, has been elected president of the association. Mr. Bannan's entire business career has been with the Western Gear Works, Seattle and its associate company, The Pacific Gear & Tool Works which organizations he now heads as president. He has given twenty-five years of service to these two companies. In addition to serving as president of these two affiliated gear organizations, Mr. Bannan is also connected currently with other well-known firms. These include president of the Webster-Brinkley Co. and the Hallidie Machine Co., and member of the board of the Pacific National Bank of Seattle.

ARTHUR E. RAYMOND has been honored for the second time this year: First, his election as president of the Institute of Aeronautical Sciences and more recently Southern California appointee of President Truman to the National Advisory Committee for Aeronautics. An outstanding designer of airplanes, Mr. Raymond holds a masters degree in aeronautical engineering from the Massachusetts Institute of Technology. He joined Douglas Aircraft Co. Inc. in 1925 as stress analyst, later becoming a designer. He became assistant chief engineer and, in 1934, chief engineer. It was under Mr. Raymond's direction that the DC-series transports were engineered and developed, the B-19 bomber was conceived and built, and the A-20 attack-bomber was produced. This bomber proved to be one of the most versatile warplanes of World War II. At the present time Mr. Raymond is vice president in charge of engineering.

HARRY R. GREENLEE with his wide background in designing automatic transmissions, clutches, drive lines and various type of special machinery is well fitted for his new post as chief engineer of L.G.S. Spring Clutch Corp. Mr. Greenlee began his engineering work with F. S. Bowser & Co. Inc., designing gasoline pumps and filling-station equipment. Then for fifteen years he designed and developed clutches and transmissions for General Motors. The following three and one-half years he spent with Detroit Gear Division of Borg-Warner on the design of various types of automatic transmissions and clutches.

Weld Penmetal Steelmesh

TO LIGHTWEIGHT METALS FOR HEAVY-DUTY WORK

Airplane Fuselage of STEELMESH Welded to Stainless Steel

Lightweight aluminum and stainless steel are used for heavy-duty work by welding STEELMESH to the sheets for reinforcement. An example is the successful application of flat rolled steel reinforced with STEELMESH for the fuselage, wings, tail surfaces and cowlings of airplanes. In a given area of STEELMESH, only 10% is solid steel (the remainder is interstices). It is, therefore, possible to spot weld this mesh to a thin sheet and secure a composite construction much stronger than a solid steel sheet of the same weight.

EXPANDED METAL MESHES

PENN METAL COMPANY, INC.

The strength imparted by STEELMESH is due to the trusslike formation of the mesh and to the fact that in the manufacture of this product, the openings are simultaneously slit and cold drawn or expanded.

STEELMESH is used in the construction of bulkheads, open partitions, insulation support, guards and ventilators, lockers, cages, truck bodies, shelving, racks, baskets, drying trays, gratings, air filters, drainers, storage bins, grille work, screening, mastic flooring, dehydrators, toys, novelty work and for many other purposes.

Write for catalog and samples.

METAL MARK
Penmetal
1869
OF QUALITY

PENN METAL COMPANY, INC.
Dept. 30, Parkersburg, W. Va.
District Sales Offices
BOSTON • NEW YORK • PHILADELPHIA • CHICAGO • DETROIT • PARKERSBURG, W. VA
LOS ANGELES • SAN FRANCISCO • DALLAS

In his connection with H. R. Krueger & Co. he worked for a year on designing special machinery. In 1940 he became associated with the Studebaker Corp. where for four and a half years he supervised the design and development of automatic transmissions, clutches, drive lines, and special military vehicles. He then joined the L.G.S. Spring Clutch Corp. as development engineer, later becoming assistant chief engineer and subsequently chief engineer, his present position.

◆

George D. Shaeffer recently was elected vice president of Gar Wood Industries Inc., Detroit. He had formerly been chief engineer of the company.

◆

J. R. Ferguson has been appointed director of automotive engineering of the Packard Motor Car Co. Relinquishing his duties as chief production engineer of the company's war output of aircraft engines, he will take up his new work in the expanded peacetime development of the Packard car.

◆

W. W. Burrows Sr., formerly assistant to the first vice president and chief engineer of Aircooled Motors Corp., Syracuse, N. Y., is now vice president and general manager of the new company Williamsport Paper Box Co. Inc., Williamsport, Pa.

◆

Col. Jesse G. Vincent, vice president in charge of engineering, Packard Motor Car Co., has been elected to the board of directors. With the company since 1912 when he joined it as chief engineer, Colonel Vincent will remain in active charge of Packard engineering.

◆

C. A. Peirce, formerly vice president in charge of engineering, Diamond T Motor Car Co., has been named executive vice president.

◆

William H. Horn has been made design engineer in the Experimental Engineering Division, Bendix Aviation Corp., Detroit. Previously he had been project engineer with Stout Research Division, Consolidated Vultee Aircraft Corp., Dearborn, Mich.

◆

Lawrence W. Fischer, Nelson R. Brownyer and **Paul V. Osborn** have recently been elected vice presidents of the Timken-Detroit Axle Co., Detroit. Connected with the engineering department since he came to Timken in 1921, Mr. Fischer in his new post will continue in an executive engineering and sales capacity. Prior to joining Timken in 1926 as sales engineer, Mr. Brownyer had been chief engineer of Gotfredson Truck Corp. Mr. Osborn became associated with Timken in 1941 as assistant to the vice president. Previously he had been general manager, and later, vice president of the Carrier Corp.

◆

Vice Admiral Emory S. Land, who served as chairman of the SAE Aircraft Engine Subdivision in 1928-1929, recently resigned as chairman of the U. S. Maritime Commission to become president of the Air Transport Association.

◆

Robert E. Kerr has joined Hercules Motors Corp., Canton, O., as engine designer. He had formerly been layout man with Chevrolet Motor Co.

◆

Richard K. Koegler is project engineer in the Helicopter Product Division, Bell Aircraft Corp., Niagara Falls, N. Y. Prior to this appointment, Mr. Koegler served as head of the design strength section of Curtiss Wright Corp. Airplane Division, Buffalo Plant.

◆

George K. Floroff, formerly design engineer with Joshua Hendy Iron Works, has joined Pratt & Whitney Aircraft, Division of United Aircraft Corp., East Hartford, Conn.

◆

W. C. Newberg, previously chief engineer, Dodge Chicago Plant, Division of the Chrysler Corp., has been appointed a staff executive of the Chrysler Corp., Detroit.

◆

Merton H. Blank, who had been chief engineer of the Hydraulic Coupling Division of the American Blower Corp., has been named vice president in charge of engineering and sales, Liquid Drive Corp. The firm, formed by Mr. Blank and an associate, will specialize in designing, manufacturing and selling of hydraulic couplings, as well as manufacturing a closed traction type of liquid coupling.

◆

Wesley A. Styer has become a designer with Northrop Aircraft Inc., Hawthorne, Calif. He had been section head of Goodyear Aircraft Corp., Akron, O.

◆

Charles A. Gladden Jr. has been named chief engineer, Gladden-Haas Inc., Owosso, Mich. Previously he was test engineer in the Aircraft Engine Division, Continental Motors Corp., Muskegon, Mich.

◆

Bernard Last, previously a designer with M. W. Kellogg Co., is now design engineer and partner in Imagineering Unlimited, Rahway, N. J., which specializes in the design and development of special purpose machines.

◆

Joseph T. Kulhavy has joined J. I. Case Co., Rock Island, Ill., as senior engineer in charge of engine design.

◆

W. S. Kreis is now body designer with Nash Motor Division, Nash-Kelvinator Corp., Detroit.

◆

William H. Peet, until recently chief engineer, Detroit Universal Duplicator Co., is now a partner and chief engineer of Die Industries, Detroit.

◆

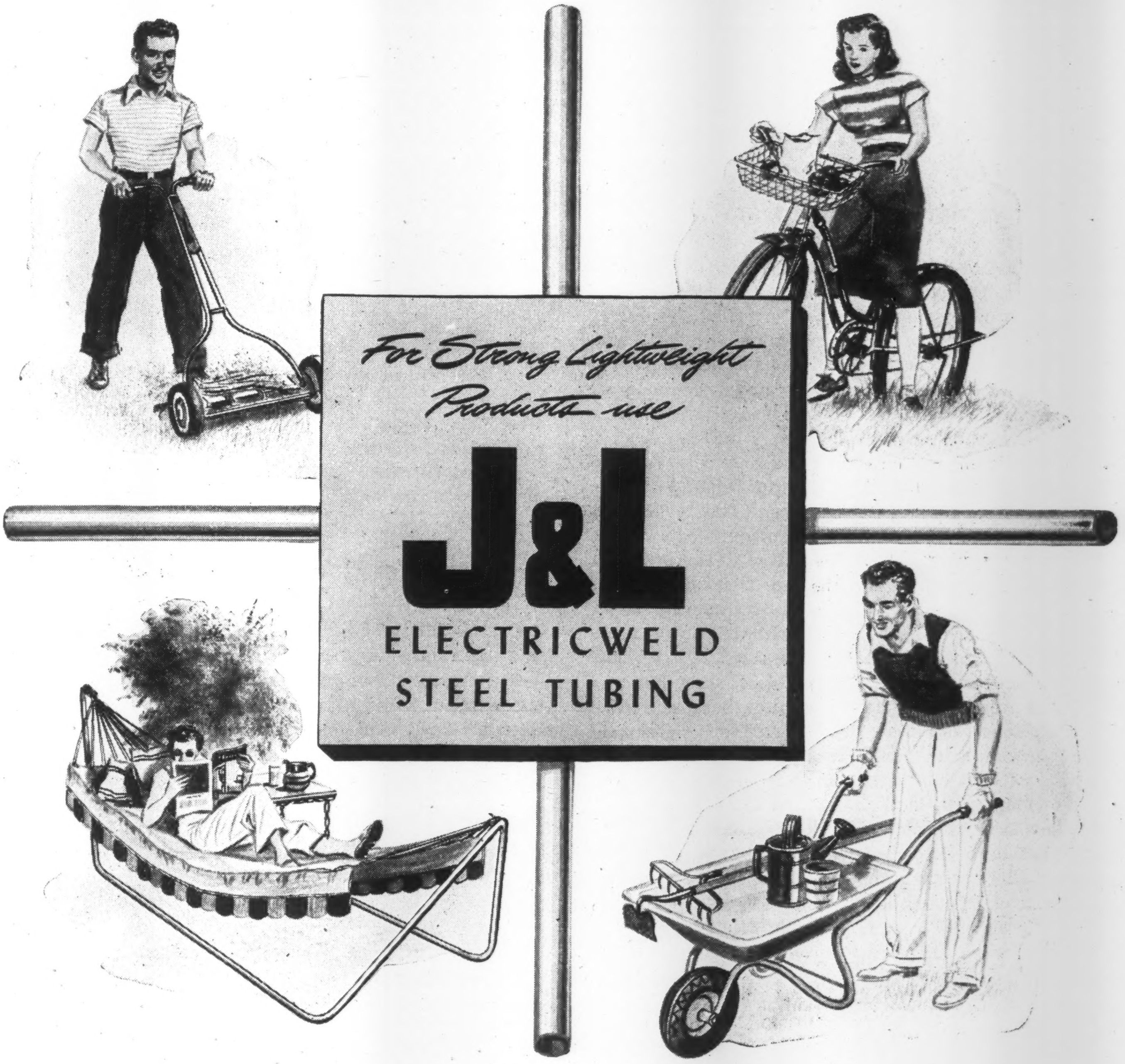
Carl R. Hennicke has resigned as chief engineer, Eberhard Mfg. Co., Cleveland, to join United Mfg. Co., Bedford, O.

◆

B. F. Zeigler Jr. has recently become connected with the Vulcan Bearing Machine Co., Dallas, Tex., as production engineer in charge of engineering and production.

◆

Hudson T. Morton as standards engineer for Fafnir Bearing Co. will have charge of coordinating engineering standards for the company. Previously he had been connected with Precision Parts Co.


For Strong Lightweight Products use
J&L
ELECTRICWELD
STEEL TUBING

Smart modern designs for garden furniture, lawn mowers, wheel barrows, bicycles and other home utilities can be enhanced by use of J&L Electricweld Tubing. Its smooth, uniform surface provides an ideal base for painting, enameling and electro-plating, and most important its inherent strength and quality permits you to build with a minimum of weight—to sell the public with ease. Also J&L Electricweld Tubing will work smoothly in your bending, fabricating and welding operations.

J&L STEEL

JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PENNSYLVANIA

Design Abstracts

Improvements in Magnesium Alloys

MAGNESIUM-BASE alloys have entered the fold of aircraft structural materials but recently. There are a number of reasons for this late arrival. Magnesium was not even produced commercially on any important scale before the first World War, and the increased production then was used largely in the nonstructural forms of ribbon and powder for flares and incendiaries. After the war, production again fell to a mere dribble. People were afraid of magnesium, and even metallurgists were not sure that it was safe to have a bar of it indoors, in the event that it should burst into flame. The tremendous use of magnesium for incendiary bombs during the recent war did not help to allay these fears. There is still a fairly prevalent belief that the use of structural magnesium entails a greater fire hazard than actually exists. Almost any metal in a finely divided form can be made to burn, and magnesium is not an exception.

Of the structural forms of magnesium, sheet has presented the greatest flammability problem, because its thin section permits localized overheating and subsequent melting more readily under certain conditions of combustion. At least for the structural forms of magnesium, actual melting of the metal is required before any burning can take place. Recently, an alloying addition to magnesium has been developed which exhibits a remarkable inhibiting effect on the burning characteristics of the metal. The action of this addition is so potent that magnesium sheet is not affected to an appreciably greater extent than steel sheet, when subjected to incendiary ignition.

Corrosion Resistance

Another obstacle that magnesium alloys have had to overcome, before becoming accepted for aircraft, is excessive corrosion. The early attempts to utilize magnesium alloys as engineering materials were not outstandingly successful because the melting fluxes then used were of such a character that they became entrapped in the metal, and carried over into the final product. These fluxes contained chlorides which would effloresce during atmospheric exposure and would cause severe corrosion of the magnesium alloys of that time. Superior fluxes and improved metal-handling methods have eliminated these flux inclusions and their corrosion-promoting effects.

For the past three years, magnesium alloys in the wrought form have been produced commercially at a very low impurity level. It was found that extremely small quantities of certain other metals, contained as impurities in magnesium alloys, exerted a marked influence on salt-atmosphere corrosion resistance. Iron is especially harmful in this respect, but copper, nickel, and silicon are also undesirable, although less frequent contaminants. By production control methods, it has been possible to keep these impurities at very low levels, so that the present-day wrought magnesium alloys are not only extremely resistant to ordinary atmosphere corrosion, but may be considered corrosion-resistant in salt atmospheres as well.

It is surpising how many otherwise well-informed people are ignorant of this improvement in magnesium alloys, and who still believe that wrought magnesium will dissolve in a few hours if immersed in sea water. Of course, as is the case when joining any two unlike metals, precautions must be taken in insulating magnesium from contact with other, more noble metals, to prevent galvanic corrosion.—*From a paper by J. C. De Haven, research engineer, Battelle Memorial Institute, presented at the recent SAE National Aeronautic meeting in New York.*

Design of Planes for Rental Service

A NATIONWIDE airplane rental service may be the next great step in American aviation development —if engineers can be persuaded to design airplanes to the specifications of those who sell, rent and operate them. Such a service would enable the customer to rent a plane cheaply and conveniently, fly to his destination, and leave the plane at another station.

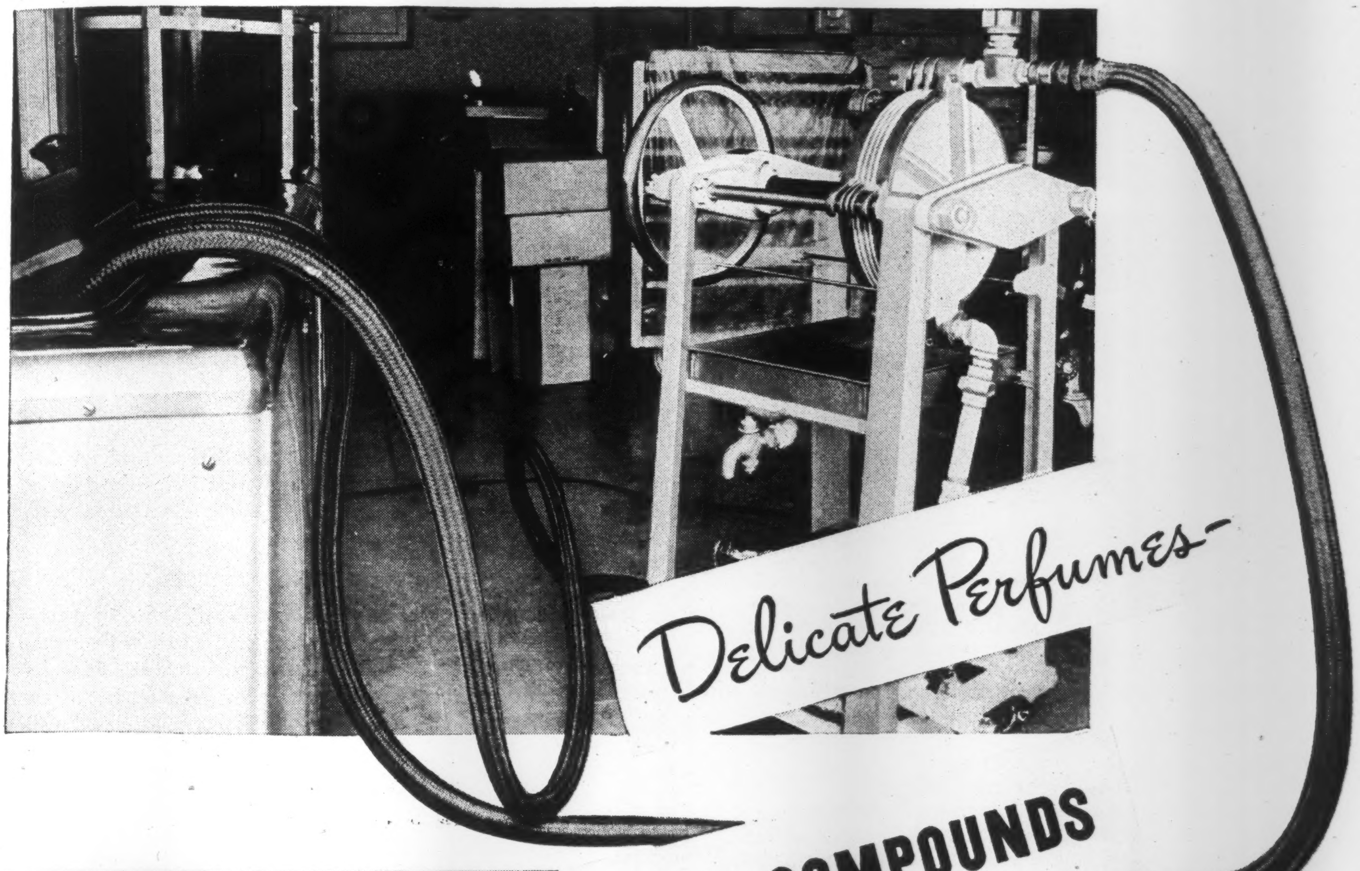
An example of what might be accomplished in the personal flying field is the projected Fly-Urself system—a cooperative undertaking to be participated in by a national organization of operators.

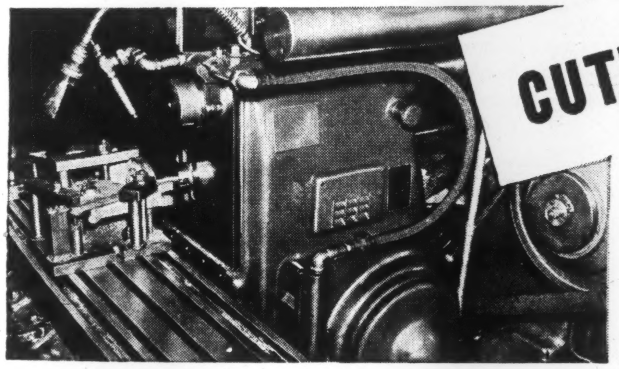
Rates will be based on a new power-time recorder, that will give not only the time the engine is in operation, but also the amount of actual power taken from the engine. Rates will be based on cruising at 60 per cent of power, so that overspeeding will be expensive.

As the vehicles for this extensive aircraft plan, it is proposed to develop two airplanes—a two-place and a four-to-five-place plane. Both are to be all metal with tricycle, retractable landing gear. The two-place plane is to have a 75-horsepower engine, weigh 1500 pounds, have a range of 500 miles, cruising speed of 125 mph, payload of 420 pounds including baggage, and useful load of 600 pounds. The four-place plane will weigh 5000 pounds, have two engines of 250 horsepower each, a range of 1000 miles, cruising speed of 200 mph, payload of 1050 pounds, and useful load of 2200 pounds. With a sufficiently large production schedule prices of $4000 for the smaller plane and $12,000 for the larger are not at all impossible.

With mass application of this system, it will be possible to smooth operations to such an extent that no overhaul will be done on the planes; parts will be replaced, even engines, so that repair work can be done in the shop while planes continue to operate. This represents a large challenge to the aircraft engineer, but not an insurmountable one. Results will justify efforts expended, in terms of employment, service, and volume of business.—*From a paper by Howard Ailor, Fly-Urself Airplane Service, presented at a recent meeting of the Detroit Section of SAE.*

• *The readily-maintained cleanliness of American Seamless assures this user complete freedom from impurities that might contaminate delicate perfume being carried between storage tank and filter.*





• *American Seamless is standard equipment on this make of milling machine. Used for carrying oils, solvents and cutting compounds, its sturdy toughness assures long, economical service.*

BOTH conveyed through *Flexible* Connections

These illustrations show but two of thousands of various applications of American Flexible Metal Hose and Seamless Flexible Tubing. For these durable products are continually being adapted to new and "different" engineering needs.

Steam, oil, water, gases, refrigerants and other fluids ... at high or low pressures and temperatures ... are safely and economically handled by American assemblies.

Whether *your* problem is one of compensating for vibration, connecting moving parts, or providing for misalignment, American has ... or can quickly develop ... a flexible assembly to do the job.

For detailed information on American Flexible Metal Hose and Seamless Tubing, write for Publication SS-50.

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ANACONDA

American
METAL HOSE
THE AMERICAN BRASS COMPANY
American Metal Hose Branch
General Offices: Waterbury 88, Connecticut
Subsidiary of Anaconda Copper Mining Company
In Canada: Anaconda American Brass Ltd.,
New Toronto, Ont.

YOUR WIFE'S HAT...

AND AN

Armature Shaft's Protection

Photograph Courtesy
Merrimac Hat Co.

have much in common!

Felt Roller Bearing Wick, showing an ingenious method of reservoir filling. Metal feed tube insert permits rapid oil replacement. While the oil is moving to the friction point, it is being filtered . . . freed from impurities.

Yes indeed, the smart Felt "creation" from the milliner's on the Avenue . . . and the humble Felt wick or pad that solves difficult machine lubrication problems are both *felted* from lamb's wool.

Nothing but Felt will hold oil ready to bleed immediately at dangerous friction points. This "watch dog" alertness of Felt results from its capillarity — makes it the preferred agent for lubricating large and small armature shafts, journal bearings, spindles and remote machine parts. Felt pads absorb, store and distribute oil, water, ink and other process fluids in automatic printing equipment, numbering and postage machines.

There are many other uses of Felt in industry — as shock, vibration and sound-absorber, thermal insulator, polisher, spacer. To obtain the benefits of technological advances in Felt, consult an American Felt Company sales engineer or write on your letterhead for appropriate Data Sheets and samples of Felt.

AMERICAN FELT COMPANY DATA SHEETS

Write for those you need to complete your Felt reference file.

- No. 1—Felt Density and Hardness
- No. 2—Adhesives for Felt Application
- No. 3—"K" Felt—Sound Absorption and Thermal Insulation
- No. 4—Special Felt Treatments
- No. 5—S.A.E. Specifications and U. S. Navy 27F7
- No. 6—Felt and Lubrication
- No. 7—A.S.T.M. Methods of Test for Wool Felt, D461
- No. 8—U. S. Army Specification No. 8-15G
- No. 9—Sheet Felts, standard Grades and Specifications
- No. 10—Vibration Isolation With Felt
- No. 11—Felt Seals, Their Design and Application
- No. 12—Flame-proofed Felt
- No. 13—Felt in Compression
- No. 14—Vistex—Packings, Gaskets, Seals

American Felt Company

TRADE MARK

General Offices: GLENVILLE, CONN.

New York; Boston; Chicago; Detroit; Philadelphia; Cleveland; St. Louis Atlanta; Dallas; Los Angeles; Seattle; San Francisco; Portland

BUSINESS AND SALES BRIEFS

FORMERLY connected with Copperweld Steel Co., Robert G. Willaman has been appointed sales manager and member of the board of directors of the newly organized Peterson Steels Inc., 420 Lexington Ave., New York 17. The company recently purchased SKF Steels Inc. and is engaged in the same type of business.

◆

Hugo S. Erickson and Carl E. Benson have dissolved their partnership to form a corporation known as Industrial Molded Products Co. Inc. It is located at 2035 Charleston Ave., Chicago 47.

◆

Recently announced by the Colgate Aircraft Corp. of Amityville, Long Island, N Y., is the purchase of the former Grumman Aircraft Engineering Corp. building located at 14th St. south of West Hoffman Ave., Linderhurst, Long Island, N. Y. Augmented facilities of the new factory will be coordinated with existing facilities at the Amityville plant.

◆

With headquarters at the general office in Trenton, N. J., James P. Stewart has been appointed by the De Laval Steam Turbine Co. to serve as special representative for centrifugal blowers and compressors. Formerly he was assistant general manager of B-W Superchargers Inc., Milwaukee, Division of the Borg-Warner Corp.

◆

A new type of construction will be used by Aluminum Co. of America in the erection of a twenty-story office building at Park Ave. & 58th St., New York. The entire facade, with the exception of the window areas, will be made of aluminum.

◆

Operating out of the factory office at 34 East Logan St., Philadelphia, Les Hunt has been named direct factory representative for Automatic Temperature Control Co. Inc. Mr. Hunt, an electrical engineer, was director of research for the past five years.

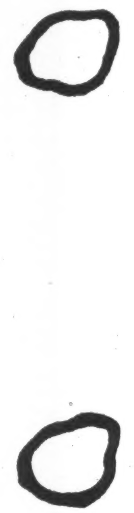
◆

Cutler-Hammer Inc. has opened a new office at 1404 North Main St., Rockford, Ill. G. I. Wolff, who has handled the territory for nineteen years, is in charge of the new office while R. E. Ford, with the Milwaukee sales organization for five years, will be resident sales engineer.

◆

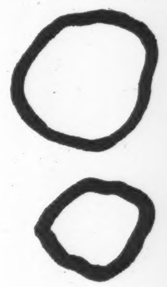
Appointment of six new field engineers has been announced by Nelson Sales Corp., Lorain, O. These are: Richard O. Blankmeyer, 211 Ledyard Drive., Fayetteville, N. Y., to cover the upstate New York and northern Pennsylvania area; Kenneth Knotts, P. O. Box 1978 or Suite 235, Executive Floor, William Penn Hotel, Pittsburgh, to handle the western Pennsylvania, southeastern Ohio, western Maryland and West Virginia area; Leslie E. Bluhm, 7762 Emerson Rd., West Lanham Hills, Hyattsville, Md., to cover the Delaware, eastern Maryland, Washington, Virginia and North Carolina area; Al

FIRST . . . LOOK AT THIS



An oscillogram, magnified 10,000 times in terms of spindle movement, showing shift of axis of lathe-spindle (3×10^{-4} in.)

THEN . . . AT THIS



An oscillogram, also magnified 10,000 times in terms of spindle movement, showing smaller shift (10^{-4} in.) after five-minute shutdown.

These Photographic Recordings, by showing effects of shutdown on lathe-spindle accuracy, helped produce precision parts in quantity.

THE PROBLEM was the mass production of small precision parts on a lathe to exceptionally close tolerances . . . and tolerances were not being met . . .

Test apparatus employing an electronic circuit in conjunction with photographic recording equipment was set up to study the performance of lathe-spindles. It proved, as shown above, that the longer the shutdown, the greater the shift of spindle axis . . .

Immediate product improvement . . . with a lower rejection rate . . . was achieved through this investigation. And this was just the beginning — further study of additional oscillograph traces revealed other irregularities in spindle performance which led to improvements in design.

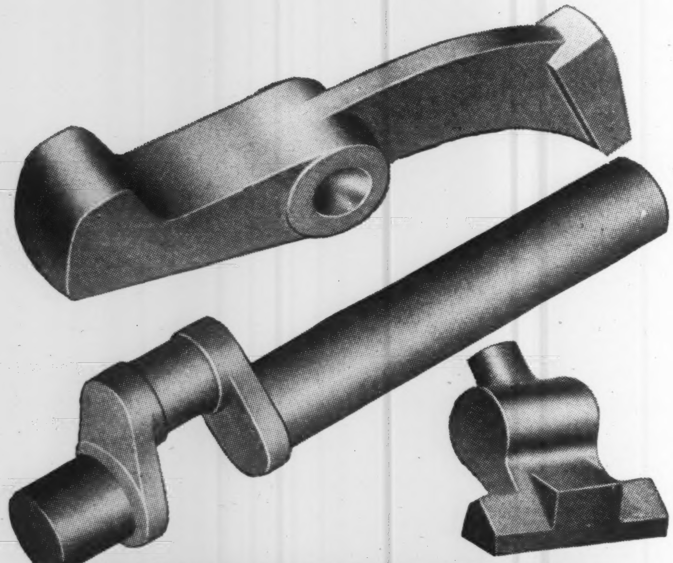
This is only *one* way in which Kodak photographic equipment, film, and paper are helping solve problems in engineering, design, production, research. For information on photographic recording materials write for the new booklet, "Kodak Materials for the Photography of Cathode-Ray Tubes."

EASTMAN KODAK COMPANY
INDUSTRIAL PHOTOGRAPHIC DIVISION
ROCHESTER 4, N. Y.

INSTRUMENT RECORDING

. . . another important function of photography

Kodak



MADE TO "TAKE IT"
THEY'RE DROP FORGED

For sheer ability to "take it" drop forged parts are unsurpassed That's why Forgings by Phoenix can be found wherever parts with a plus amount of stamina are required . . . parts that must be strong, tough and possessed of high fatigue resistance . . . parts that must not fail despite high speed operation, reversal of stresses, and continuous wear and abrasion.

Strength and stamina, however, are just two of the advantages provided by the use of Forgings by Phoenix. Others include: the ability to position strength without increasing weight . . . the forming of parts to close tolerances which minimizes machining time and eliminates waste . . . and the wide range of welding adaptability for the fabrication of complicated assemblies from two or more forgings.

The many advantages of Forgings by Phoenix are a definite aid to product improvement . . . increased efficiency . . . and production economies. You can enjoy these same advantages . . . consult a Phoenix Forging Engineer today . . . there's no obligation.

Forgings by
PHOENIX

Forging Division of
PHOENIX MANUFACTURING COMPANY
CATASAUQUA, PA. • JOLIET, ILLINOIS

Wrigley and Jack Godley, 2970 West Grand Blvd., Detroit, to handle jointly the Detroit area, with Mr. Wrigley also covering northeastern Michigan and Mr. Godley, northwest Ohio and the Fort Wayne, Ind., territory; and William G. Tawse, Box 469, Los Angeles, to handle the southern California area formerly covered by M. E. Tougas.

◆

Now under construction is a new addition to the plant of the Lincoln Engineering Co., St. Louis. To be completed by midsummer, the new addition will give relief to the demand for lubricating equipment and allied products.

◆

Opening of a New England sales office has been announced by Fawick Airflex Co. of Cleveland. It will be under the direction of Stanley T. Johnson, care of Transmission Engineering Co., 102 St. Mary's St., Boston 15. Mr. Johnson has had experience with General Electric, Electric Storage Battery and Bellamy-Robie Inc.

◆

Raymond L. Seitz and Irving C. Maust have been appointed to the West Coast sales engineering staff of Foote Bros. Gear & Machine Corp., Chicago, and will be located at the office in Pasadena, Calif. Serving as sales engineers, Mr. Seitz and Mr. Maust will assist West Coast manufacturers in the application of Foote Bros. "A-Q" (aircraft quality) gears, power units, actuators and other power transmission equipment.

◆

Associated with the company for nineteen years, George T. Walne has been promoted to the position of manager of the central district sales territory of General Box Co., Chicago.

◆

Seven field representatives have been appointed to as many district and branch sales offices of SKF Industries Inc. They are: J. K. Redding, Philadelphia office; L. E. Jacobs, Buffalo; W. S. Corneille, Milwaukee; F. O. Hamlet, Chicago; G. E. Mayhew, Cleveland; B. J. Fraher, Detroit; and C. L. Eberlein, Atlanta.

◆

Stewart-Warner Corp. has appointed Charles I. Kraus as sales manager and Gustave Treffeisen as assistant sales manager of the Alemite distribution division. Formerly Mr. Kraus was head of the industrial sales division, while Mr. Treffeisen was in charge of automotive and farm sales of equipment and lubricants.

◆

A new West Coast Division plant in Los Angeles will be built by Square D Co. for the manufacture of control and electrical distribution equipment. This will replace the plant now in operation, expanding present facilities by sixty per cent.

◆

Appointment of the following has been made by Ilg Electric Ventilating Co., Chicago: E. J. Stone, manager of the Detroit office to succeed A. H. Kirkpatrick; W. L. Hochschild, sales engineer at the Chicago branch office; John D. Briggs, assigned to the branch office in Philadelphia; and William S. Gorham, sales engineer in the branch office in Los Angeles.

◆

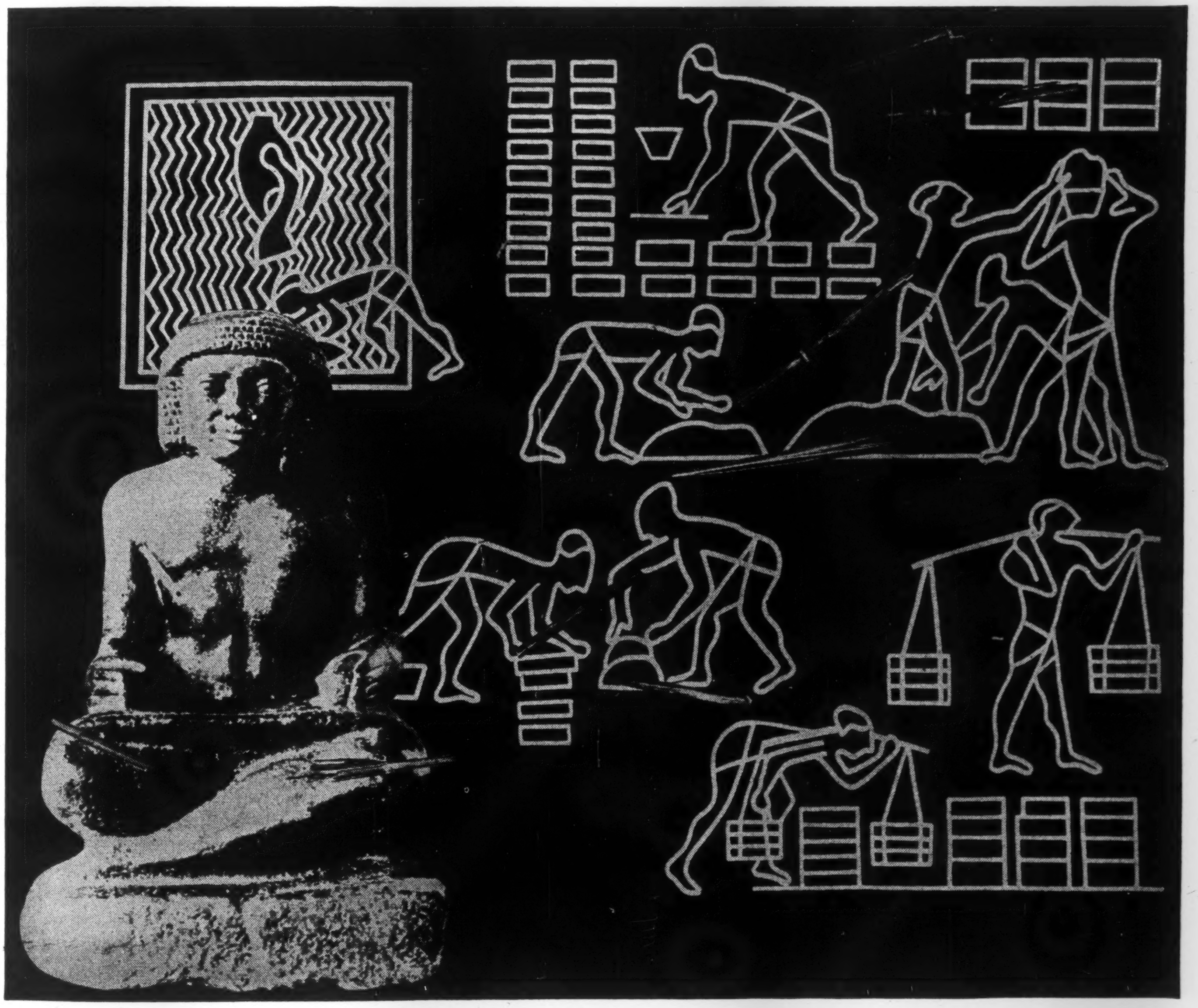
Previously superintendent of the Valve & Bender Division of The Parker Appliance Co., Cleveland, Henry M. Reese has been made special field sales application engineer. He

BRICKS WITHOUT STRAW

Long, long ago a penny-conscious Pharaoh tried to cut his cost corners by decreeing that bricks would be made without straw. He found out—the hard way—that scrimping on materials throws product performance and customer good will into full reverse.

Hardenability in steel is analogous to straw in bricks. Leave it out and you invite trouble in. Molybdenum steels combine good hardenability with economy, thus insuring good performance on a practical cost basis.

The Pharaoh may possibly have lacked data on the importance of straw in bricks—but there is readily available to all steel users a wealth of practical facts on the advantage of molybdenum-containing steels. Investigate *now!*



MOLYBDIC OXIDE—BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"
CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

Climax Molybdenum Company
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CHIKSAN

Straight flow

FIRE MONITOR

For *complete, fool-proof* fire protection, install CHIKSAN *Non-Spin* Fire Monitors at strategic locations. Easy turning is provided by double rows of ball bearings, permitting the Chiksan Fire Monitor to be turned full 360° in both horizontal and vertical planes. Monitor can be locked in any desired position. Since the discharge flow is in the same plane as the supply line, the Chiksan Fire Monitor will not spin. This means greater safety to the operator. Nothing to tighten or adjust. The Chiksan Fire Monitor is ready for instant action all the time. Write for latest catalog.

OTHER USES:
Hydraulic Washing Equipment, Log Debarking, Mixing Equipment. Can be supplied with any type of nozzle required.

CHIKSAN REPRESENTATIVES IN PRINCIPAL CITIES
EXPORT REPRESENTATIVE: Chiksan Export Co., Brea, Calif.
BRANCHES: New York 7, Houston 2

CHIKSAN
TRADE MARK REG.
BALL BEARING SWIVEL JOINTS FOR ALL PURPOSES
CHIKSAN COMPANY
BREA, CALIFORNIA
New York 7 Houston 2

will continue to make his headquarters in Cleveland but will devote all his time to consultation on valve and fitting problems of major industries, including chemical process, power plant, refining, machine tool and refrigeration.

◆

Establishment of a new district office in Cleveland has been announced by Pressed Steel Tank Co., Milwaukee. With Edward Elliot Jr. in charge, the new office will cover Ohio, most of West Virginia, the western portions of Ontario, New York, Pennsylvania, Maryland and Virginia, and the eastern portions of Michigan, Kentucky and Tennessee. Mr. Elliot recently returned to the company after his release from the Army Air Forces.

◆

Charles U. S. Grant has been appointed to the Cleveland office of Foote Bros. Gear & Machine Corp. In his new position as sales engineer he will assist manufacturers in their power transmission problems.

◆

A new company known as Resistors Inc. has been organized by Joseph J. Cerny, formerly president and general manager of Lectrohm Inc. which he founded in 1937. Located at 2241 Indiana Ave., Chicago 16, the company will manufacture a complete line of resistors as well as a wide range of rheostats, individual small-capacity solder pots, metal resistor cages, radio-frequency and power-line chokes, and custom-made heating elements.

◆

Change of name to Hammond Mfg. Corp. has been announced by the Pacific Fabricating Co. D. C. Hammond will continue as president.

◆

General offices of The Emerson Mfg. Co. have been moved to 8100 Florissant Ave., St. Louis 21.

◆

Malcolm C. Durbin has become associated with Chicago Metal Hose Corp. and will serve as staff project engineer for the company, which has manufacturing plants at Maywood and Elgin, Ill.

◆

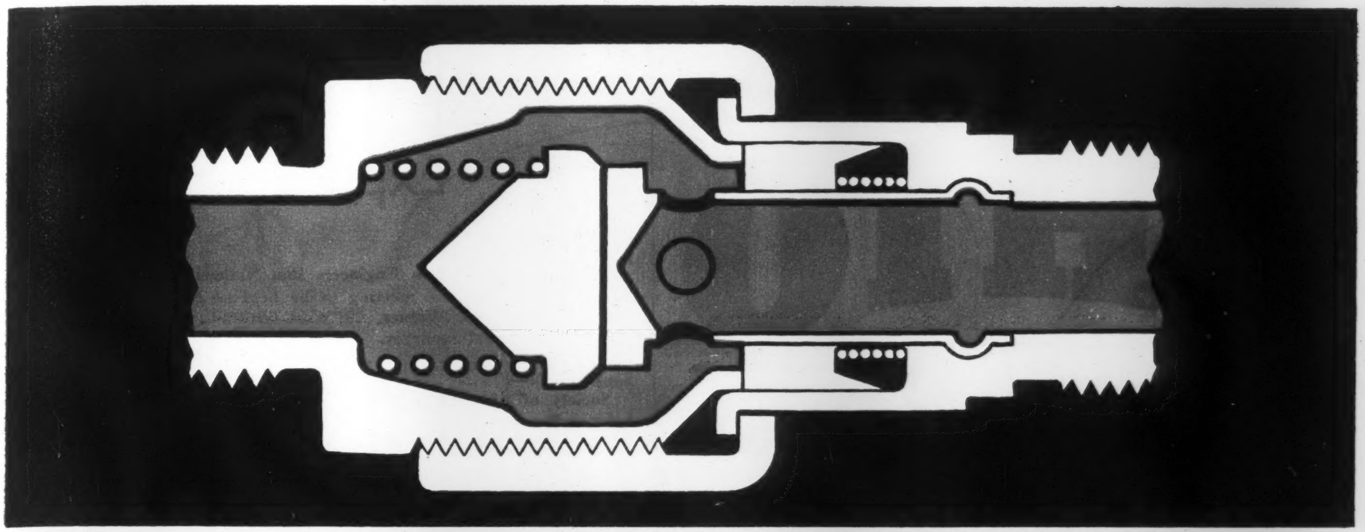
With offices in Los Angeles, E. J. Sanders has been appointed western manager of the Thermo-Aire Division of Evans Products Co., Detroit. Assisting him will be William A. Lippman, formerly with the Sky Products Division of the company. Frank Dolinich has been named assistant to J. J. Brandon, eastern sales manager of the Thermo-Aire Division, with offices at Cranford, N. J., while Frank A. Chase has been appointed central sales manager of this division, with offices in Cleveland.

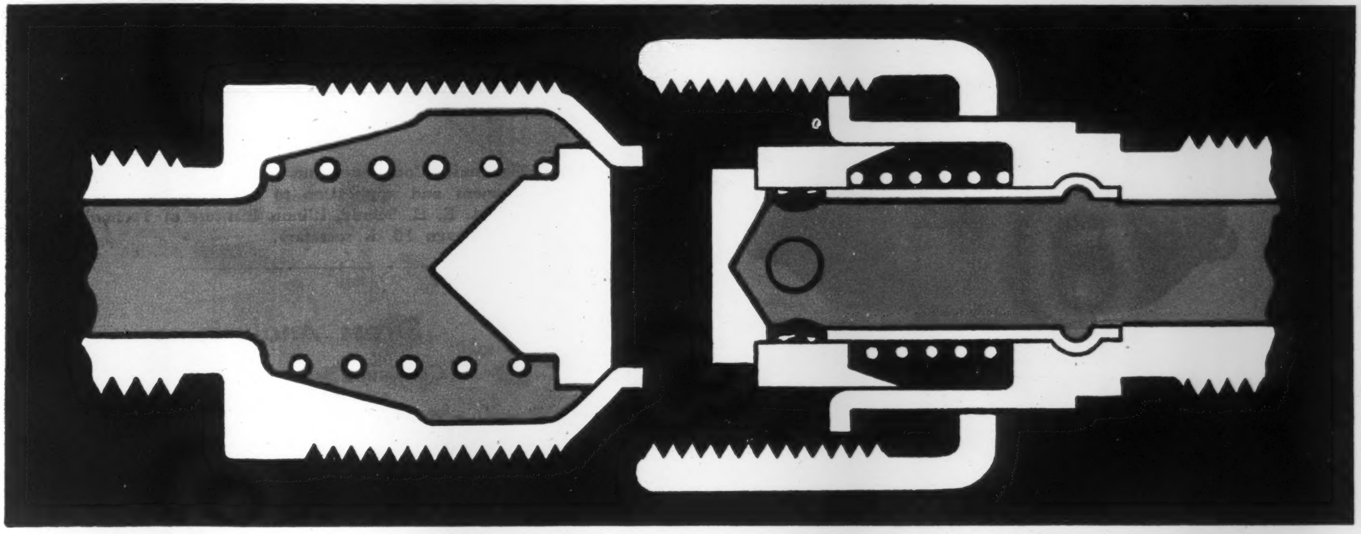
◆

Operated for the past five years as a partnership, the Chester Mfg. Co. of Lisbon, O., has been incorporated as the Chester Hoist Co. Hal F. Wright, founder of the business, is president.

◆

Atlas-Pacific Engineering & Distributing Co. has moved into its new main plant on Hollis St., between 66th & 67th Streets in Emeryville, Calif. The company specializes in the design and production of equipment for the food processing industry. In addition to manufacturing its own lines of equipment, the company serves as distributor for American Pulley Co., U. S. Electrcial Motors Inc. and Waukesha Foundry Co.





AEROQUIP Self-Sealing Couplings*

eliminates the use of 2 Valves

With the Aeroquip Self-Sealing Couplings, fluid carrying lines may be instantly disconnected without leakage. Upon separation each half of the coupling automatically closes the line or tube to which it is connected and positively retains the fluid carried therein. No leakage or loss of fluid occurs and no air is allowed to enter the system on reconnection. Suitable for hydraulic and refrigeration systems as well as any lines carrying oils, fuels, coolants, or refrigeration gases.

AEROQUIP FIRSTS

Detachable Hose Fittings
Hydrofuse
X-Washers
*** Another Aeroquip First**

AEROQUIP CORPORATION
JACKSON, MICHIGAN

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OHIO
SMALL MOTORS

for General Purpose Applications

Types—Split Phase, Capacitor, Shaded Pole, and Direct Current
Speeds—1100, 1700, 3400 RPM
Power—1/80 to 1/20 HP, depending on speed
Voltages—up to 440 A.C. and 250 D.C.
Frequencies—60, 50, and 25 cycles
Bases—Solid, Resilient or Flange
Mountings—Horizontal, Sidewall, Ceiling or Vertical, with shaft up or down
Efficiency—High for such small powers

OHIO MOTORS INCLUDE

Split-phase induction to 1/3 HP	Torque to 100 oz. ft.
Direct Current to 1 HP	Synchronous to ½ HP
Capacitor to 2 HP	A.C. to D.C. Motor Generators to 500 watts
Polyphase to 2 HP	High to Low Voltage D.C.
Aircraft Motors	Dynamotors up to 300 watts
Shell Type to 7½ HP	

What is your problem?

THE OHIO ELECTRIC MFG. CO.
5906 Maurice Avenue
Cleveland 4, Ohio

Meetings and Expositions

Aug. 22-24—
Society of Automotive Engineers Inc. National West Coast transportation and maintenance meeting to be held at New Washington Hotel, Seattle. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Sept. 11-12—
Society of Automotive Engineers Inc. National tractor meeting to be held at Hotel Schroeder, Milwaukee. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Sept. 16-20—
Instrument Society of America. First national instrument conference and exhibit to be held at William Penn Hotel, Pittsburgh. Richard Rimbach, 1117 Wolfendale St., Pittsburgh 12, is executive secretary.

Sept. 30-Oct. 2—
American Society of Mechanical Engineers. Fall meeting to be held at Boston. Additional information may be obtained from headquarters of the society at 29 West 39th St., New York. C. E. Davies is secretary.

Oct. 3-5—
National Electronics Conference Inc. National forum on electronic research, development and application to be held at Edgewater Beach Hotel, Chicago 14. E. H. Schulz, Illinois Institute of Technology, Technology Center, Chicago 16, is secretary.

Stress Analysis

(*Continued from Page* 136)

is a small radius present in all practical gages, and this is what contributes most of the lateral sensitivity. In stress gages ($2\phi \approx 60$ degrees) this radius causes the angle to deviate from the theoretical value only slightly, the jig design being adjusted to produce a stress gage of the proper value of Poisson's ratio.

The gage factor for stress gages must necessarily be expressed in terms of stress rather than strain, and since the gage design restricts the gage operation (with respect to Poisson's ratio) the material upon which the gage can be correctly used is also restricted. Consequently, the expression for gage resistance variation with stress that corresponds to the familiar strain gage equation

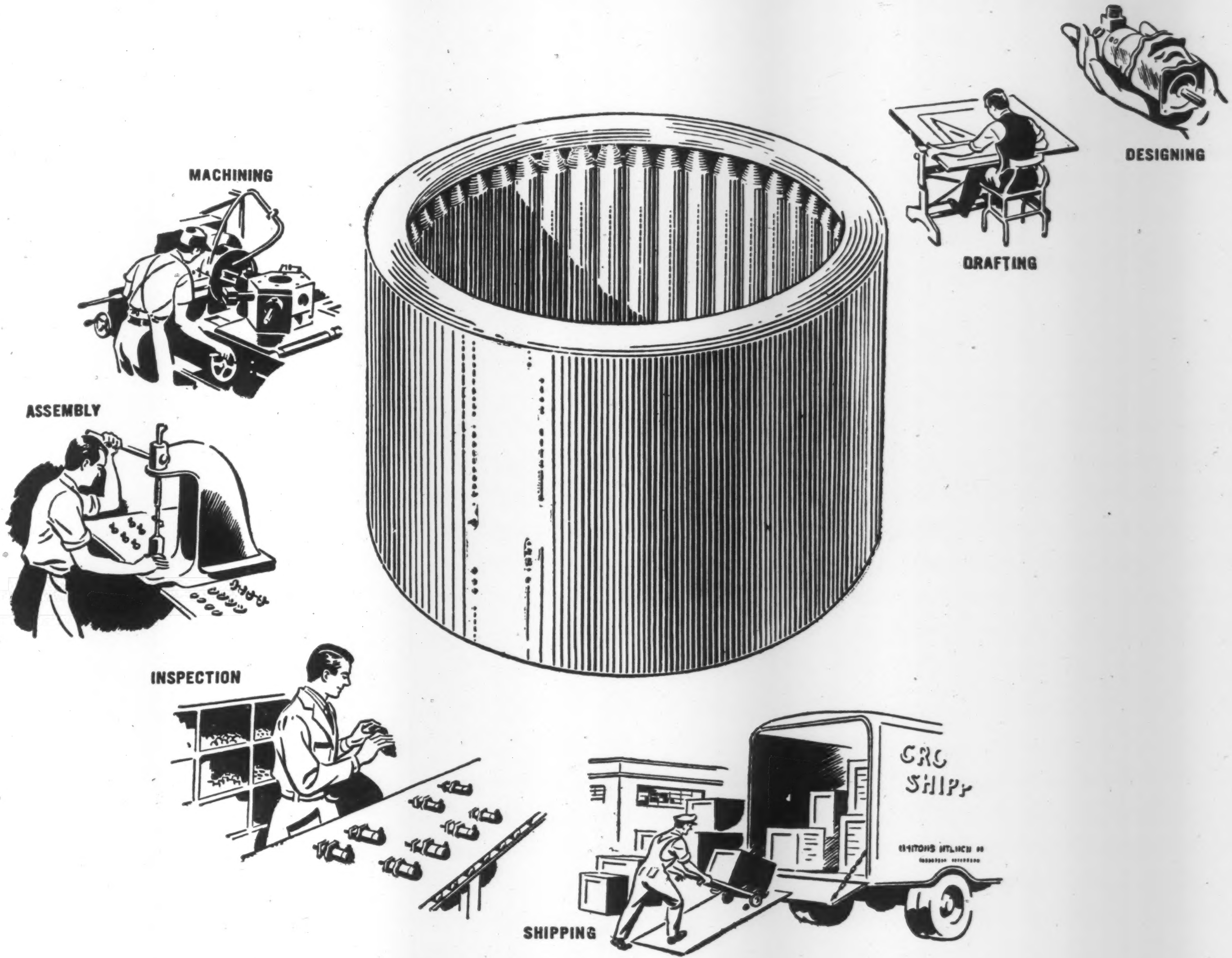
$$\frac{\Delta R}{R} = S\epsilon \qquad (9)$$

is

$$\frac{\Delta R}{R} = F\epsilon' \qquad (10)$$

where R is the gage resistance, S is the strain gage factor, F the stress gage factor and ϵ and ϵ', respectively, the strain along the gage axis and the ratio of stress along the gage axis to Young's modulus for the material upon which the gage is mounted. These latter two quantities are associated with the resistance changes ΔR. For the stress gage, the material upon which the gage is mounted must possess the Poisson's ratio for which the gage is designed.

It develops that $F = (1-\mu)S$. This is explained by the following reasoning: Since a stress gage has an output of the general form of Equation 8, it can be considered to be a gage with an L-shaped wire-path, such that the



Here's Anti-Friction Performance for Assembly Lines, too...

One thing about using Torrington Needle Bearings that the plant men appreciate is the way they can help to speed up the efficiency of assembly lines.

It's not only that their *unit construction* simplifies handling and speeds installation. The advantages start right back in the design stage where their small size and light weight help to solve many related design problems.

Then getting ready for production is easier, too. No elaborate tooling-up is necessary—a plain bore, machined to proper dimensions, provides the bearing housing.

Installation is a simple press fit...a single arbor press operation sets the bearing in position. No spacers or retainers are required to hold Needle Bearings in place.

And because of the efficient lubrication of this modern anti-friction unit, special or complicated systems for lubrication are never required. Even oil seals are unnecessary in many applications.

So, when you adopt the use of Needle Bearings, you can count on increased efficiency of your own production lines. And that's something we're all looking for these days.

Our engineering department will gladly assist you in securing full benefits from the use of Needle Bearings.

THE TORRINGTON COMPANY
TORRINGTON, CONN. SOUTH BEND 21, IND
Offices in All Principal Cities

TORRINGTON NEEDLE BEARINGS

BRIGGS & STRATTON ENGINES

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shorter leg is μ times the length of the longer leg (which latter must lie along the axis of stress measurement). Consequently, the output for uniaxial stress is less than that of a strain gage made of the same piece of wire arranged in one straight path (which approximates closely the usual strain gage). It follows from Equation 7 and the fact that $\tan^2 \phi = \mu$ that

$$\frac{\Delta R}{R} = \frac{S}{2}(\epsilon_{+2\phi}+\epsilon_{-2\phi}) = S\frac{\epsilon_x+\mu\epsilon_y}{1+\mu} \quad \text{.......................(11)}$$

But, from Equations 8 and 10,

$$\frac{\Delta R}{R} = (1-\mu)S\frac{\sigma_x}{E_y} = F\epsilon' \quad \text{.......................(12)}$$

From the definition of ϵ'

$$(1-\mu)S\epsilon' = F\epsilon' \quad \text{.......................(13)}$$

yielding

$$F = (1-\mu)S \quad \text{.......................(14)}$$

which means that the output of the stress gage is less (for simple axial stress) than a similar strain gage in the ratio of $(1-\mu)/1$.

Thus, C-1 strain gages usually possess a gage factor $S = 3.5$. CV28-1 stress gages (similar to C-1 strain gages but arranged for measuring stress in a material with Poisson's ratio of 0.28) have a gage factor of $F = 2.5$ or 28 percent less. They may be checked by mounting them transversely to the loading direction under simple axial stress, under which condition their output is negligible.

Stress Field Need Not Be Biaxial

It should be noted that the advantages of stress gages over strain gages lies wholly in the field of biaxial stresses, but since the stress gage will operate faithfully in either uniaxial or biaxial stress fields, its general use eliminates the need for establishing whether the gage to be used is likely to encounter a loading for which it will not respond accurately. Inaccuracies in *stress* measurement due to strain measurement by a single gage may have any value, since the strain may be zero in some directions for which the stress may be significant. If it is assumed, however, that knowledge of the loading is always sufficient to locate the strain gages in the direction of the larger principal stress, the relative error in assuming $\sigma_x = E_y \epsilon_x$ will lie between $+\mu/(1-\mu)$ and $-\mu/(1+\mu)$ as σ_y ranges between $\pm\sigma_x$. To date, no stress gages have been made with any other resistance than 500 ohms nor will a gage length appreciably less than that of the C-1 gage.

SR-4 stress gages, types CV28-1 and CV28-2 (the latter has the shorter mounting centers) have been in quantity use for some time in the measurement of vibratory stresses in hollow steel propeller blades, *Fig.* 1, where the case of simple or uniaxial stress distributions is the exception rather than the rule. Stress gages exhibit a marked advantage when measuring complex vibratory stresses because the biaxial stress equations are solved continuously without recourse to a second measurement plus electrical or mental calclulation. This is essential in propeller work especially, for the slip-ring system severely limits the number of gages which may be used in any one flight.

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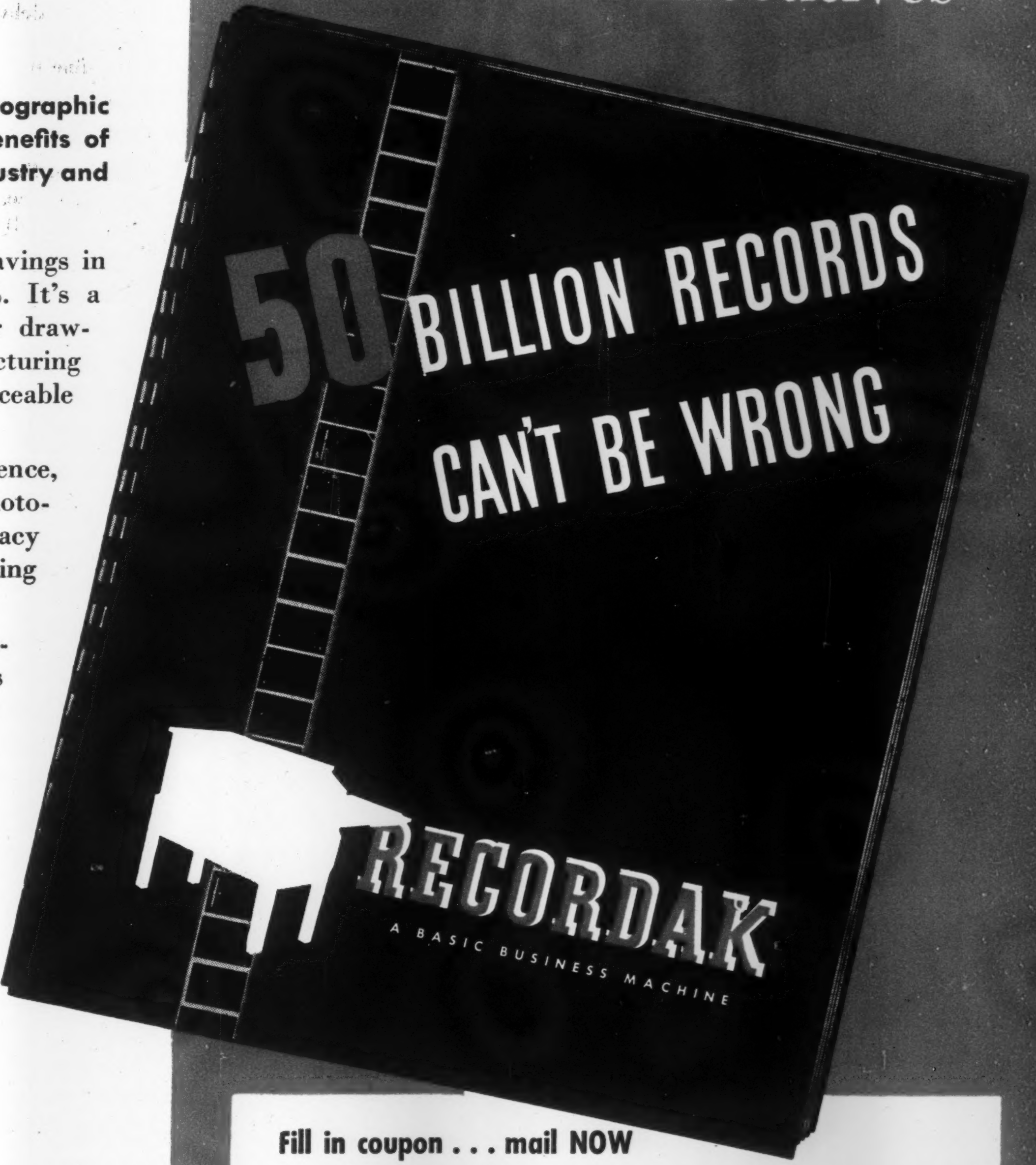

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